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Heating · Piping and Air Conditioning

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VEMBER, 1939

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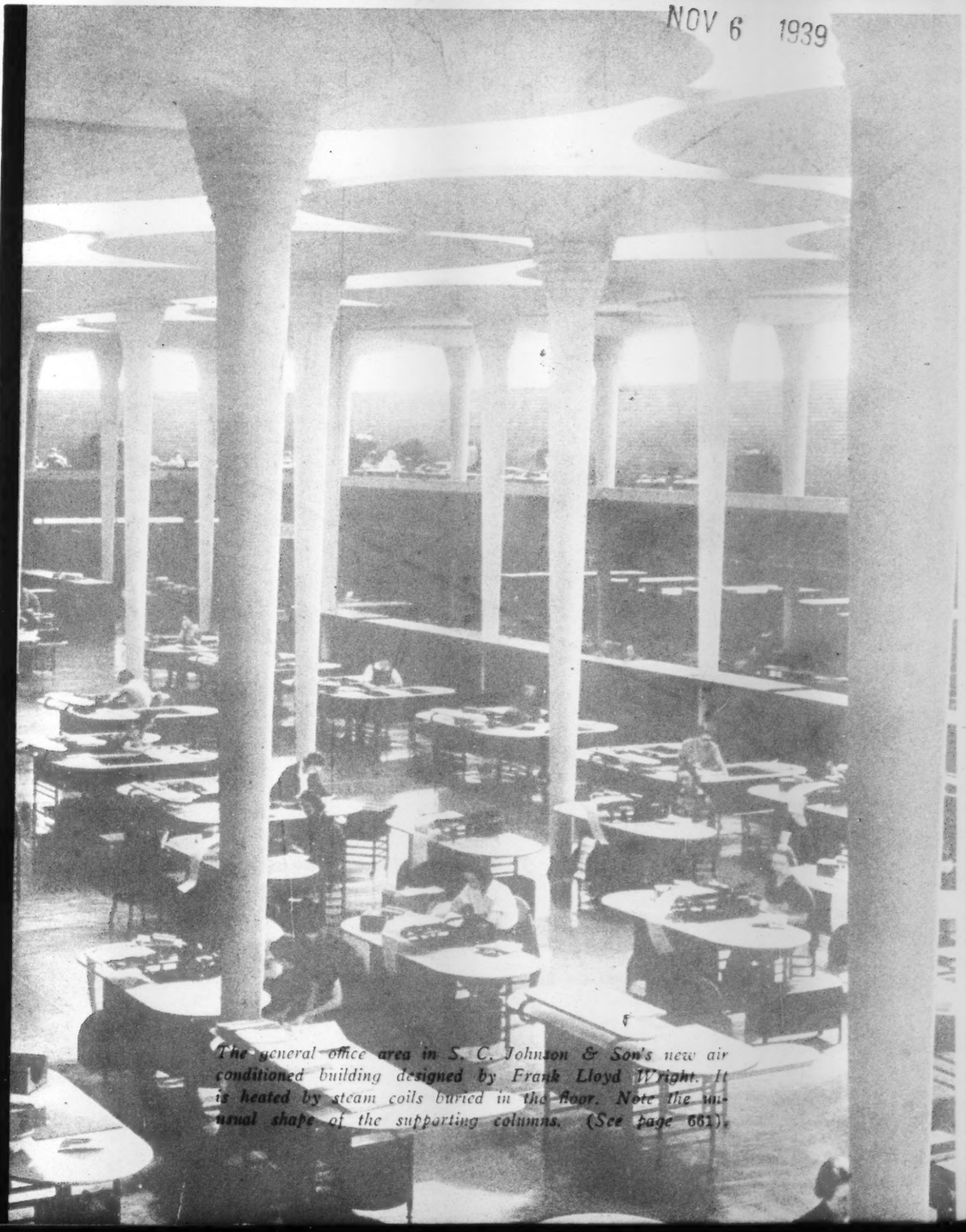
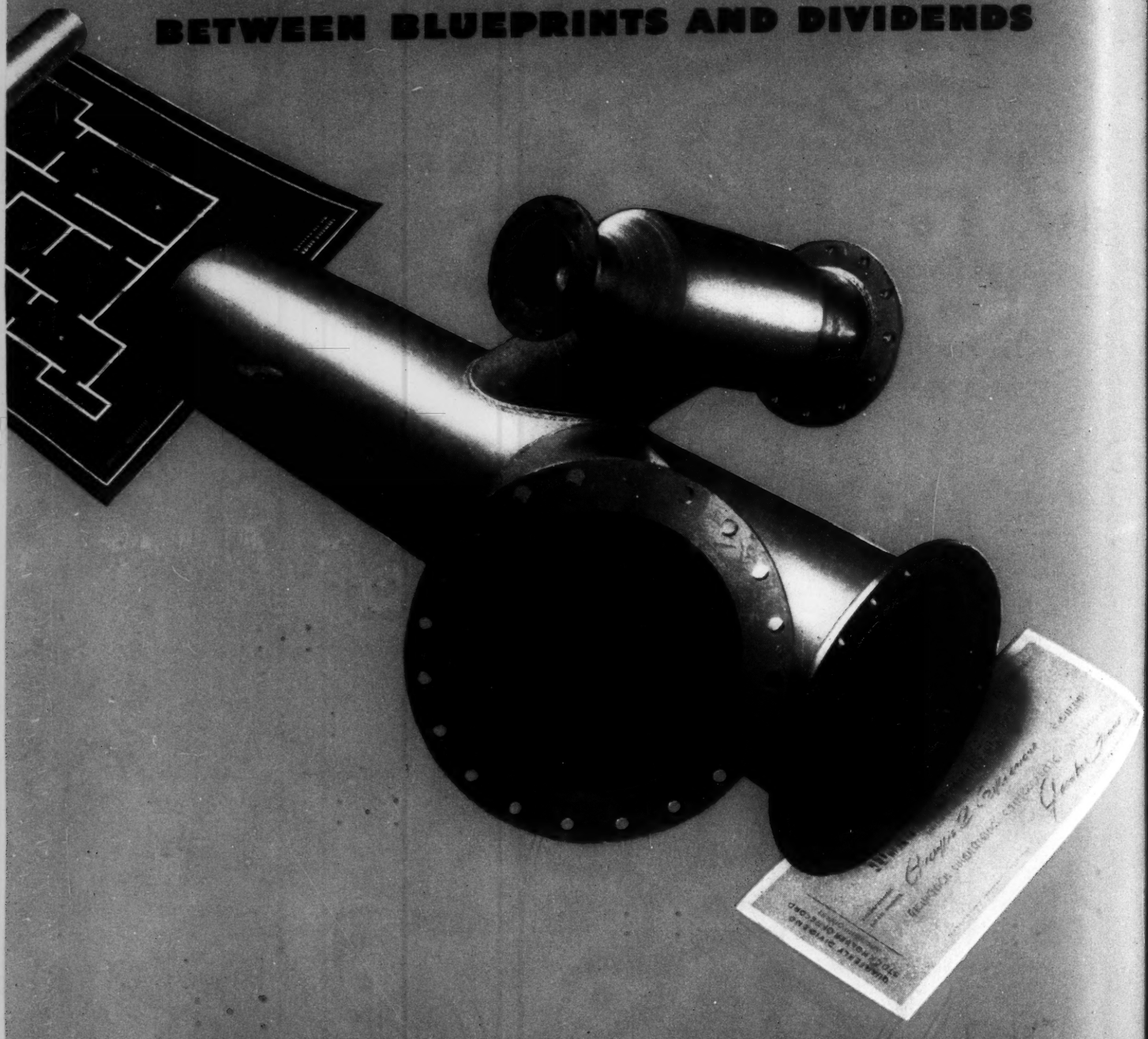


The general office area in S. C. Johnson & Son's new air conditioned building designed by Frank Lloyd Wright. It is heated by steam coils buried in the floor. Note the unusual shape of the supporting columns. (See page 661).

Shortest Bridge

BETWEEN BLUEPRINTS AND DIVIDENDS



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WHENEVER **PIPING** IS INVOLVED

THE EDITOR'S PAGE

COOLING TOWER ASSOCIATION FORMED

L. T. Mart, Marley Co., has been elected president of the newly formed Water Cooling Tower Association. W. J. Hoffman, Lilie-Hoffman Cooling Towers, Inc., is the vice-president and the secretary-treasurer is W. J. Parker, 7 E. 44th St., New York, N. Y.

Among the plans of the association are a standardization program and the establishment of a code of ethics.

CENSUS TIME IS HERE

Enumeration of the 1939 census of business is to begin on January 2, and is scheduled for completion next June, according to the Bureau of the Census of the Department of Commerce. This is the fourth business census and the third in which the construction industry has been included. At least the basic facts learned from it are to be available late in the summer of 1940.

An answer to the age-old proposition of "how the other half lives" will be available when the first nation-wide housing census is concluded next year, according to the U. S. Housing Authority; it is to be made in conjunction with the 1940 census of population.

SHORT NOTES AND COMMENT

Our front cover features again this month an air conditioned windowless building—that of the makers of Johnson's Wax, in Racine, Wis. Another interesting thing about this building (which is described in our lead-off article) is the unusual heating system. . . . Air conditioning has made possible the construction of windowless buildings. Among outstanding structures of this kind are the Detroit Edison Co.'s new service building, the National Aluminate building, the Simonds Saw and Steel plant and office building, and the Hershey Chocolate Corp.'s building. The heating and air conditioning of all of these modern structures have been described in articles published in these pages. . . . Windowless buildings are to be discussed at the production conference of the American Management Association, November 15-16, Palmer House, Chicago.

A public hearing on the code for dust separating apparatus recently formulated by an ASME committee has been scheduled for Monday, December 4, at 4:30 p. m., in the Bellevue-Stratford hotel, Philadelphia. Copies of the proposed code may be obtained from ASME headquarters, 29 W. 39th St., N. Y.

Awards of honorary life membership in the (British) Institution of Mechanical Engineers to two members of the ASME, A. G. Christie and Henry Ford, were reported last month. Professor Christie was the author of an article published in the May, 1939, HPAC on high pressure and temperature piping progress during the past 10 years.

Declaring that the future of our children depends on their health and education, and the future of our country depends on our children, Dr. Albert G. Young, medical director of Corey Hill hospital, has proposed that elementary schools be provided with air conditioning. The measure was suggested at the International Association of Electrical Leagues meeting in New York City.

AIR CONDITIONING BOOSTS OCCUPANCY

Occupancy of four office buildings in New Orleans, La., rose from 72.4 to 87 per cent after air conditioning was installed, according to a report issued recently by a group of realtors in cooperation with Carrier Corp. The study was made over a 10 year period. The four buildings are the American, Whitney, Canal Bank and Hibernia.

The study showed that these buildings reached in the period from 1933 to 1938 an average percentage of occupancy higher than the previous all-time high of 86.9 per cent in 1929. Just prior to the installations occupancy had dropped to a low average of 72.4 per cent. During the period from 1929 through 1938, non-air conditioned buildings in New Orleans dropped from an average percentage of occupancy of 91.2 per cent to 67.2, according to the report.

BROOKINGS INSTITUTION HEAD ANALYZES INDUSTRIAL POLICIES IN WAR EMERGENCY

The economic goal of the United States at this time should be to make a full utilization of our economic resources in expanding production without permitting a general inflation of prices and a destruction of equilibrium in the economic system, according to Harold G. Moulton, president of the Brookings Institution, and a member of the war resources board. Speaking at a General Motors luncheon in New York City last month, he warned that war in Europe "raises economic and financial problems of wholly unprecedented importance for the entire world."

Basing his analysis on the assumption that the United States will not become a participant in the European war, he said that economic developments on the North American continent in the coming months may well prove of decisive importance to the whole future of industrial civilization. Although available data are far from adequate, he said there is much reason for believing that the great upsurge in raw material buying which has occurred since August has resulted in a greater price advance in many raw materials than is warranted by the fundamental supply and demand factors in the situation, and that much speculative buying has been based upon exaggerated notions about world wide inflation. There is no *scarcity* basis for a great advance in the general level of commodity prices, he said.

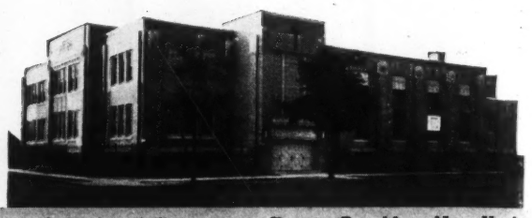
So long as an industry is operating well below capacity, an increase in volume, which distributes the burden of overhead over a larger number of units, will offset large increases in direct operating expenses. Industry should be content, he said, to make increased profits by a fuller and steadier utilization of capacity, and by ever-increasing efficiency in production.

RMA PUSHES EQUIPMENT STANDARDS

The Refrigerating Machinery Association stressed need for equipment standardization in the refrigerating machinery industry at its recent annual meeting. Several proposed standards were reviewed, particularly those affecting performance of vertical, single acting ammonia compressors. Plans for the immediate future include completion of studies dealing with "Freon-12" compressors, mechanical condensing units (ammonia and "Freon-12"), evaporative condensers, and "Freon-12" water coolers.

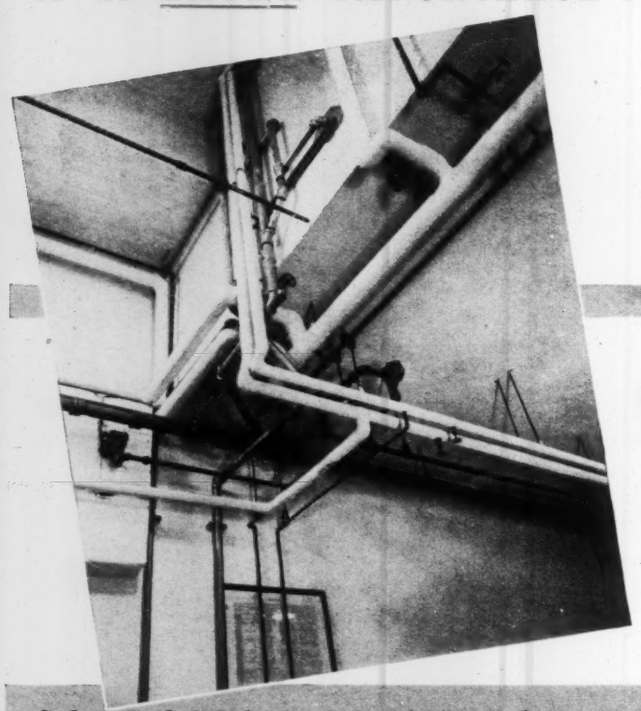
W. S. Shipley, York Ice Machinery Corp., was re-elected president of the association, and J. I. Lyle, Carrier Corp., was re-elected vice-president. The executive committee comprises D. Norris Benedict, Frick Co., H. A. Feldbush, Worthington Pump and Machinery Corp., J. M. Fernald, Baker Ice Machine Co., G. A. Heuser, Henry Vogt Machine Co., and Emil Vilter, Vilter Mfg. Co.

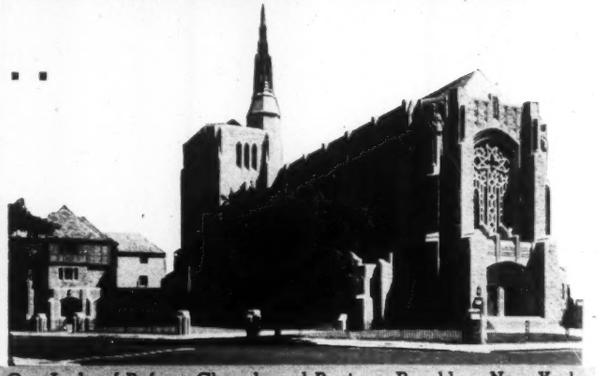
Where MINIMUM MAINTENANCE



St. Athanasius School and Community Center, Brooklyn, New York

is of EXTRA IMPORTANCE . . . .





Our Lady of Refuge Church and Rectory, Brooklyn, New York

BYERS WROUGHT IRON

Henry V. Murphy
Architect,
Brooklyn, N. Y.

CORROSION COSTS YOU MORE THAN WROUGHT IRON

Careful selection of materials for corrosive services assumes even greater importance in church and school work, where there is no operating income to take care of maintenance. Henry V. Murphy, who specializes in ecclesiastical architecture solves the problem by making extensive use of wrought iron. The interior view above shows piping in the School and Community Center of the Roman Catholic Parish of St. Athanasius, Brooklyn, New York, where specifications call for wrought iron in all waste, vent and leader lines; gas piping; roof connections; and underground return lines.

The variety of materials offered today, and the conflicting claims, make selection a highly technical and demanding job. Many architects find a service which we render of real help, and we believe you, too, will find it of value. If you will write, outlining details of your particular corrosion problem, our Engineering Service Department will (1) Determine the probable corrosive conditions involved; (2) Compare these with similar or parallel service conditions encountered elsewhere, and recorded in our files; (3) Interpret the findings in the light of 75 years' experience with corrosion problems; and (4) Make recommendations . . . supported by service records. There is no cost or obligation. Ask, too, for a copy of our bulletin, "Wrought Iron in Piping Systems."

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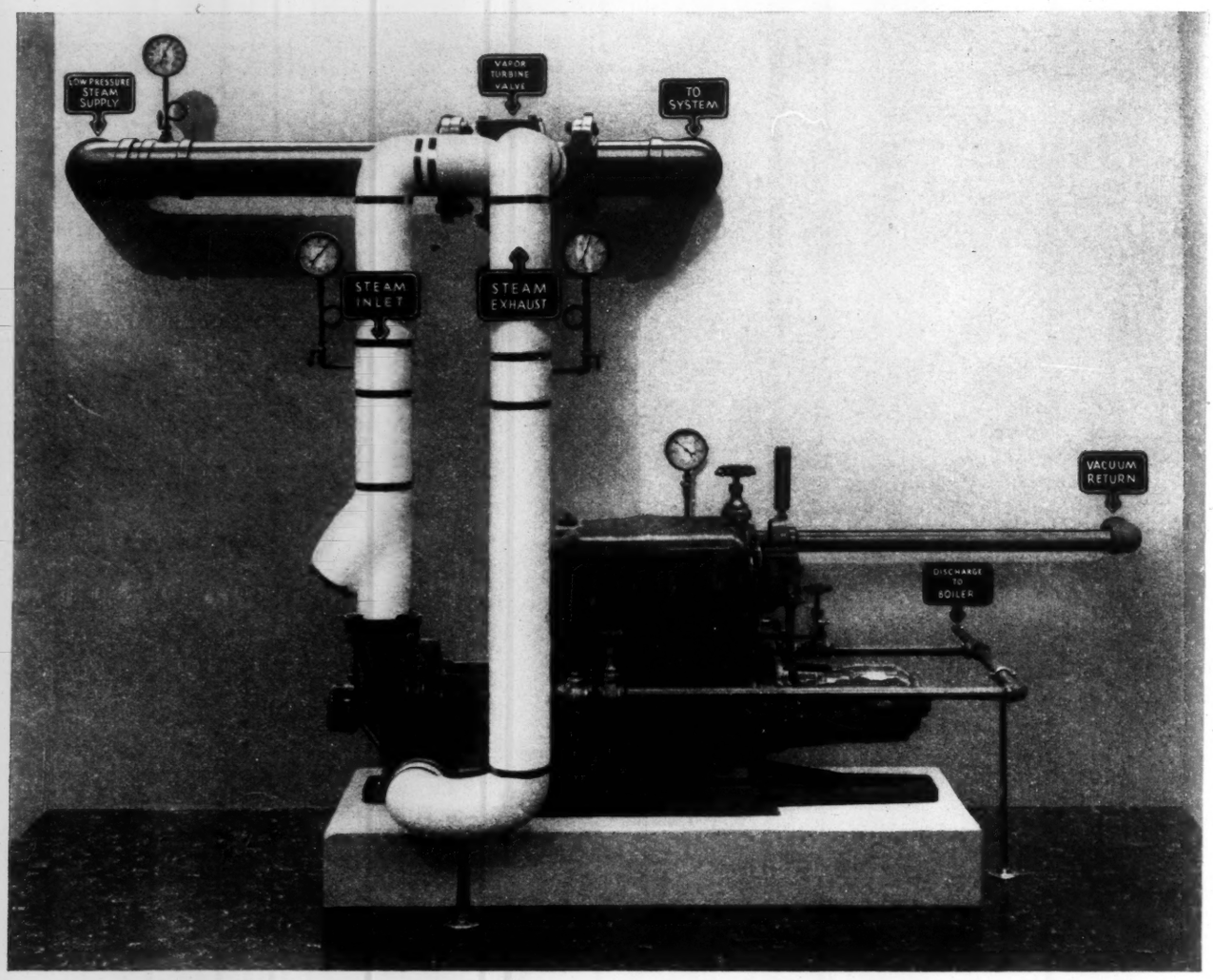
Where Heat Must Not Fail— Install this SAFE Heating Pump!



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THE NASH ENGINEERING COMPANY

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Heating · Piping and Air Conditioning

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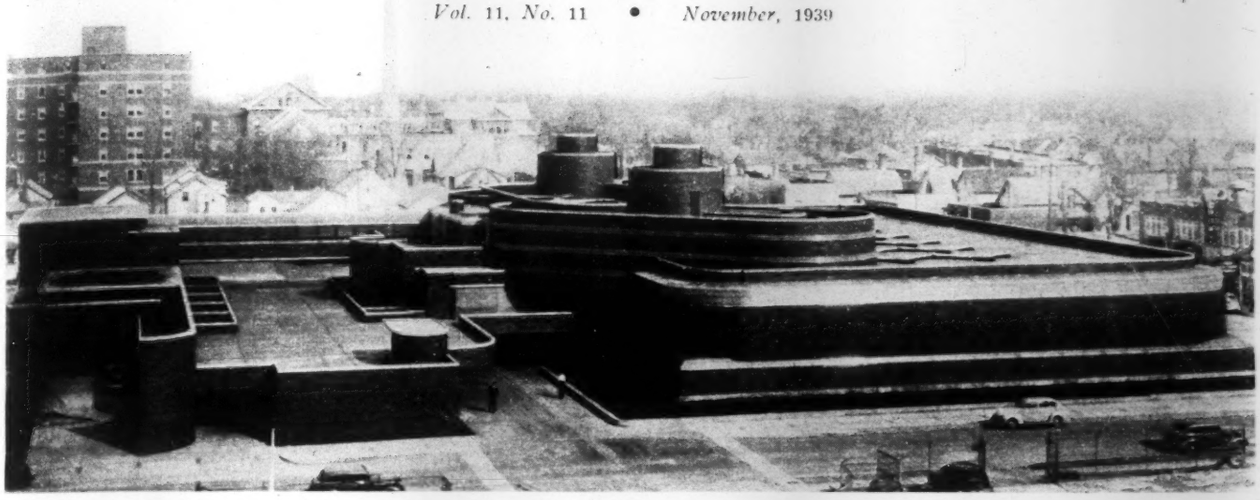


S. C. Johnson & Son's ultra-modern office building, showing the enclosed court entrance and the "nostrils" which house the dehumidifiers and serve as air inlets and mixing chambers

Building Designed by Frank Lloyd Wright Features Floor Heating, Air Conditioning

By Victor Walters*

AT Racine, Wis., stands the ultra-modern office building recently completed for S. C. Johnson & Son, Inc.—epitome of the logic and functionalism in architecture of which Frank Lloyd Wright, its designer, is the untiring champion. Though every adjective expressing fantasy has been applied to the features of construction and appearance, its famed architect has proved his reasoning by the extraordinary utility of the structure.

Basically the windowless office building presents a primary problem in air conditioning and heating, and since the design of the Johnson building called for windowless construction, Mr. Wright developed his plans with several unique angles. Inevitably air conditioning becomes a fundamental consideration in modern building planning although the Johnson building embodies features which may have a vital effect on the distinct trend toward windowless construction.

First among the unique features is the heating system for the general office which involves the principle of "permeation" rather than convection or conduction; in more common phraseology, this method is called floor heating. This type of system was used because it was believed that conventional heating methods could not be adapted to the construction details of this office, which is a large workroom 210 by 130 ft with a 20 ft ceiling.

The use of 3 in. cork insulation for walls and roof resulted in minimum heat losses while the entrance to the building from an enclosed court and driveway further reduced this factor. Though a windowless building,

In the unique office building recently completed for S. C. Johnson & Son, Inc. (Johnson's wax) at Racine, Wis., pipe coils in the concrete floor slab are used for heating. The building is essentially one large workroom, and it was felt that this method of heating was more adaptable to it than would be a more conventional type of system. Designed by the famed architect Frank Lloyd Wright, the structure has many novel features and is completely air conditioned. . . . . This description supplements an earlier preliminary article which was published in the April, 1938, HEATING, PIPING AND AIR CONDITIONING

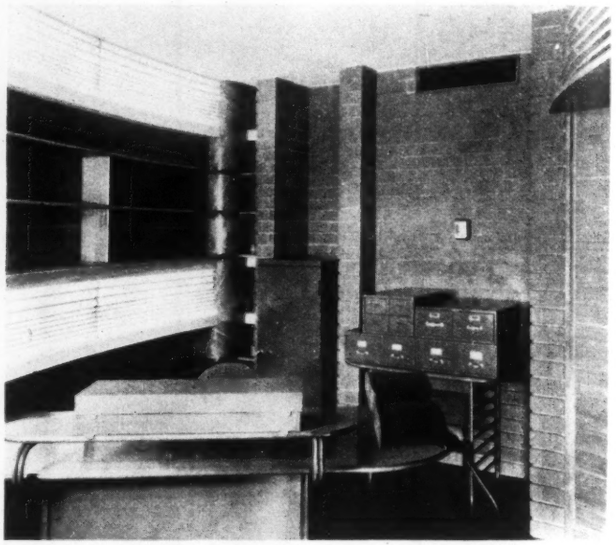
*Sales Engineer, Westerlin and Campbell Co.

Entrance hall showing tubular enclosures and walls through which light is admitted. Note also the air distribution grilles in the mezzanine enclosure

daylight is admitted through bands of clear glass tubes encircling the structure, so that the radiant effect of the sun is also considerable. The coefficients of heat transmission, the temperature of the heating medium, and the required surface area were assumed and all other design factors were based on these assumptions. Steam was selected as the heating medium because the company's nearby power plant could be utilized, while it was felt the maintenance of vacuum conditions in the system would reduce pipe corrosion. Since it was desirable to hold temperatures comparable to those in a hot water system, sub-atmospheric steam was specified.

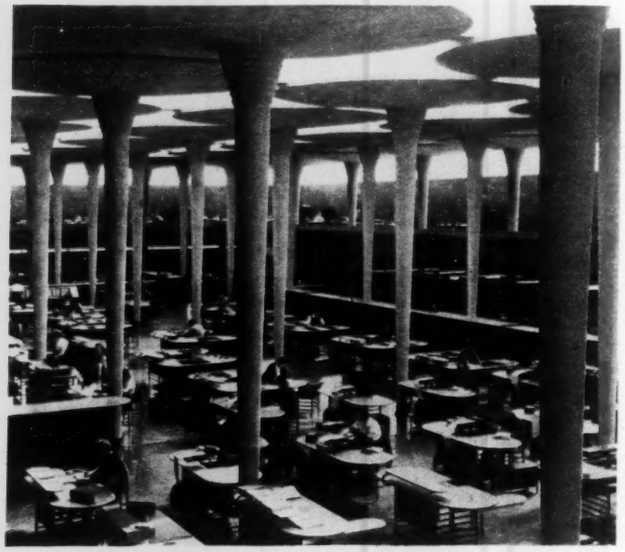
Actual installation of the heating equipment began with the placing of wood blocks, at proper grade, under the piping. The wrought iron pipes were then field welded and clamped to precast concrete blocks. The piping was then covered with coarse gravel, about 5 in. thick below the pipes and 3 in. above. A 6 in. concrete slab was poured over the gravel and finished with felt and rubber tile.

Steam at 75 lb pressure is piped from the power plant, reduced to 5 lb at the first stage, and through a second manual valve to sub-atmospheric pressure which may be varied from 0 to 28 in. vacuum. Each of the five control zones of the floor piping system is equipped with two thermostatic devices connected with a central panel which keeps conditions averaged, by operation of a steam valve, to maintain a slab temperature of 60 to 75 F.



Typical private office on the third floor of the building. This view shows clearly the glass tubes through which light is admitted, the air distribution grille, and the control thermostat

Interior view of the general office area. This space is heated by means of the floor radiation system while the ventilating and conditioned air distribution is accomplished through grilles in the wall surrounding the balcony

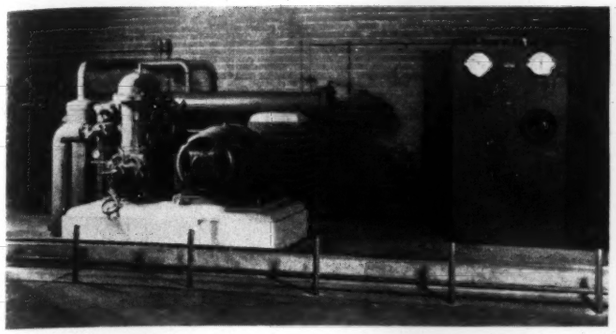


Of particular interest was the architect's assumption that the heating method would result in a substantial reduction in heat loss by comparison with convectional methods since the heat warms the building material rather than passing through it. In addition, he anticipated the air temperatures required for complete comfort will be 10 F less than usual.

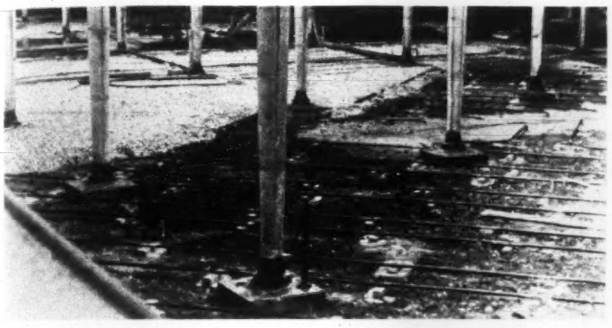
The executive offices are heated by booster coils placed in the ventilating ducts and controlled by modulating thermostats. Tempering and reheating coils in the ducts, thermostatically controlled, provide 75 F air for ventilating while the air washers are arranged to furnish humidification during the heating season.

The cooling phase of the comfort cycle is slightly less revolutionary than the floor heating system. That it contributes as much to the efficient functioning of the offices has been acknowledged by the management after a season's experience in the structure.

The air conditioning specifications for the cooling phase called for 80 F dry bulb and 50 per cent relative humidity when outdoor conditions of 95 F dry bulb and 75 F wet bulb were recorded. Compensating control



The condensing system which serves one of the two complete zones, providing complete summer air conditioning for the entire building



Construction view of the early stages showing the floor heating piping arrangement and fill

maintains the relationship between conditions outdoors and within the conditioned space in conjunction with automatic air mixing bypass devices and complete accessory equipment.

The duct work for air distribution is of the built-in type, and the ducts are part of the building structure, with framed outlets provided with manual friction type louvre dampers and special deflection grilles.

This is literally a "breathing" building with two "nostrils" extending above the roof level which serve as air inlets and house the fans and washers. The incoming air is cooled, dehumidified, and mixed in these chambers and then circulated through the structure.

Two complete systems were installed, each with its own condensing system, using a motor driven, vertical, single acting "Freon-12" compressor, and arranged for direct expansion of the refrigerant in coil type dehumidifiers which are situated in the nostrils. This arrangement of equipment was adopted because of its flexibility and to take advantage of operating economies available with zoning. Each condensing unit has a capacity of 50 to 57½ tons and is powered with a 60 hp slip ring motor with manual starter.

The fans have sufficient capacity for circulating 260,000 cfh of conditioned air; this air is introduced through specially designed outlets at a comparatively low level without noise or drafts. Return air is delivered to the dehumidifier chambers through built-in masonry ducts similar to those provided for the supply plenum.

Performance of System

Although the building has been occupied only since last May, the heating and air conditioning systems have been operated since October, 1938. Thus a complete cycle has been observed and the success of the design confirmed.

With a floor temperature of 72 F, the building was maintained at comfort zone conditions, the five control zones functioning in perfect coördination. The air distribution system has been balanced, and the original air volume calculations exceeded slightly in actual practice.

Though the cooling load was increased substantially by the installation of lighting of almost twice the design wattage, the summer air conditioning system, with an occupancy load, functioned satisfactorily.

The air conditioning and floor heating sytems were engineered and installed by the Westerlin and Campbell Co. Frank Lloyd Wright, in addition to his services as architect, assumed personal supervision of construction, which was handled by B. E. Wiltscheck, the general contractor.

[Photographs courtesy York Ice Machinery Corp.]

OWNER CITES ADVANTAGE OF AIR CONDITIONING

We have just finished the season of operating our air conditioning system on the cooling cycle. There is no doubt in our minds that during the summer months this can be credited with a large percentage of increased production over buildings without cooling. All department heads report that as a result of air conditioning (1) machine operation errors have been reduced, (2) absenteeism has been lessened, (3) heat weather fatigue has become negligible, (4) as a result of controlled temperature and humidity, the daily production of work is nearly constant regardless of outside conditions, (5) it is possible for employees to maintain a better appearance, (6) employees put in more nearly a 100 per cent working day.

The new office building is completely sealed from the outside and all air entering the building passes through air washers, resulting in an easier job of maintenance and a healthier place in which to work. During the hay fever season many of our employees hated to leave the building, for during working hours they found great relief due to the air conditioning.

—S. C. Johnson & Son, Inc.

Bank Vaults Cooled

The United States National Bank of Omaha, Nebraska, has recently installed a 5 ton air conditioning system for cooling its safety deposit vault and individual booths as well as the securities vault on the second level of the sub-basement, according to Natkin & Co. Conditioned air is supplied to the interior of the securities vault by a hinged duct which is swung out of position when the vault is sealed at night.

Each vault has a separate air conditioning unit installed outside of the conditioned space in each case. A 5 hp condensing unit and refrigerant cooled motor is installed in the machinery room of the sub-basement. This unit supplies the refrigerant to separate air conditioning coils in each air conditioning unit.

The two air conditioning units provide cooling, dehumidifying, filtering, air circulation and ventilation. Also, by an ingenious arrangement, each unit is connected to the present building air supply and is able to furnish winter conditioning as well through the same distribution system.

Piping and Handling Heavy Fuel Oils for Boilers and Industrial Furnaces

By Paul R. Unger*

INDUSTRIAL and process furnaces present in one respect a different aspect than do boilers; the furnaces are generally scattered over a large area and are generally not laid out for the convenience of the oil piping system. It has been common practice to run one main oil circulating line through the plant and take individual take-offs from this line to the various oil burners. A relief valve is usually situated at the end of this line and circulation is then continued back to the storage tank. This system keeps the main oil circulating line in an excellent temperature condition for efficient burning but does not necessarily provide sufficient hot oil to the various burners; this deficiency has caused more difficulty in burning heavy fuel oil than any other single factor. These individual take-offs may be from 10 to 100 ft in length. Many industrial furnaces require from 5 to 15 or 20 gph of oil. If we assume that the average industrial furnace will use 10 gph of oil, the average dead end run of pipe from the main circulating line to the burner is 30 ft, and the size of the individual take-off line is 1 in., we will have lying in this line nearly 1½ gal of unheated oil when lighting off. For perhaps 10 minutes, at this rate, cold oil will have to be burned or drained off.

This is a condition usually impossible to contend with if heavy oil is burned. Many plants, confronted with this problem, shift to a lighter oil because of difficulties encountered and conclude that it is impossible to burn heavy oil in this type of application. It is not only the circulating problem which is difficult, but the heating problem, due to the fact that when small quantities of oil are burned in each furnace and the only oil moving through the take-off and individual burners is that actually consumed in this particular burner, the movement of oil is very slow. The temperature drop between the heated oil circulating line and the oil burner itself is often as much as 40 to 50 F.

Let us consider a typical plant—that of the Phoenix Mfg. Co., consisting of about 15 small furnaces scattered over a distance of about 800 ft. The furnaces were burning from 5 to 10 gph of a No. 3 oil, similar to that used in heating houses. They had sometime before tried to use heavier fuel but without success. There was one oil circulating line that ran down the full length of the plant, with right angle take-offs to the individual furnaces. Asked whether it would be possible to burn heavy fuel oil in these furnaces, we examined the burners and decided that they would burn heavy oil satisfactorily without change if the oil were brought to each burner at all times at a temperature of not less than 190 F.

We constructed a large pumping, heating, and straining set. The oil temperature was controlled so that the oil would leave the pump set at a temperature of 210 F; from here the oil lines took a zigzag course running within 6 in. of each burner on each furnace. The entire line was insulated with 85 per cent magnesia. We studied the oil temperature conditions and advised the installation of a second steam operated fuel oil heater about 400 ft from the first pump set to boost the oil temperature again from 190 back to 210 F. (The temperature drop during the first 400 ft was approximately 20 F). This boost maintained the oil temperature above that required for the balance of the circulating line. As we passed the last burner we installed an adjustable relief valve set to maintain the correct oil pressure in the line and from there ran the circulating line back to the oil storage tank. With this system, it was possible to light any burner instantly and to be sure that hot oil was always available (see drawing). As a matter of fact, when this installation was completed, the burners actually operated with greater satisfaction and with less fuel on heavy oil at the correct temperature than they had with the original piping system using a domestic oil.

In plants where it is impossible or impractical to run the entire circulating line past each burner, a take-off may be used to care for one or more burners in a different portion of the plant, provided the take-off is treated similarly to the main line; in other words, run directly by each burner and then brought back and connected into the main return line between the relief valve and the oil storage tank or taken directly back to the tank. It may also be convenient to situate the fuel oil heating, pumping, and straining equipment centrally with respect to the various groups of furnaces and radiate several circulating lines from it. In each case, however, each individual circulating line must run close to each burner on its loop and then head into the return line. Where the lines are extremely long, it may be that more than one booster heater will be required. Where steam is not available at the location of the booster heater, immersion type electric heaters with thermostatic control may be used.

*Mid-West Heat Service Co., Combustion Engineers.

A timely and practical discussion of methods of piping and handling the heavier fuel oils—and even pitch—for firing industrial process furnaces is presented here. Particular attention is given to combination installations, where both boilers and process furnaces are fired with the same grade of fuel oil, using the same pumping, heating and circulating system . . . In the October HPAC, Mr. Unger discussed low and high pressure steam and forced hot water boiler plants

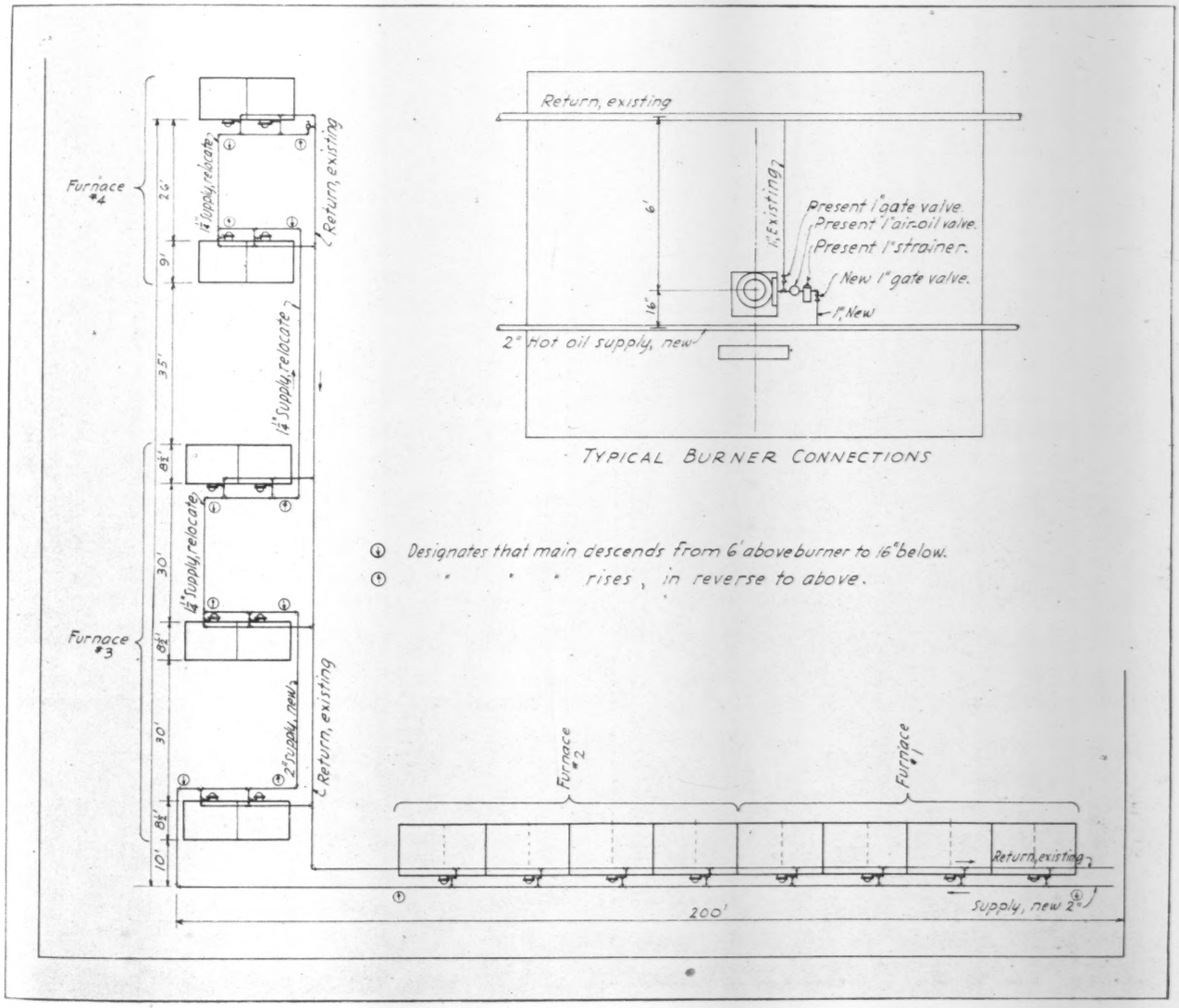
Combinations of Boilers and Process Furnaces

There are many plants which have a combination of both boilers and industrial process furnaces to be fired with the same grade of fuel. The majority of these plants maintain separate storage and separate oil handling equipment for the boilers and for the process system. It is possible—and in many cases advisable—to build one complete, well designed, duplicate pumping, heating, and straining system of sufficient capacity so that when operating one pump, one heater, and one strainer, the requirements of the burners on the boilers and the burners on the various furnaces will be adequately and properly served. This may be done even though the burners may be of widely different types and may require varying fuel oil pressures. But the same principle of circulation must be followed.

An installation recently completed at the plant of the Northwestern Malt Co. consists of three 150 hp boilers operating at 10 lb pressure (two equipped with fully automatic burners and one with a semi-automatic burner) and 18 additional burners operating direct fired malt kilns. The last burner is approximately 1000 ft from the storage tanks. The plant was operating with No. 5 oil which was heated in the boiler room to approximately 150 F. This plant had the common fault of a circulating line running along the ceiling the full length of the plant, with individual dead end take-offs to the various burners. We found in this case that the oil heaters were too small to handle Bunker C fuel oil, the oil pumps were running at 1200 rpm (too fast for handling heavy fuel oil efficiently), and that the oil storage tanks did not have heating coils. Even with No. 5 oil there was great difficulty in starting the oil burners as in winter the exterior of the firing chamber of a malt kiln approximates outside temperature, the firing room being open to outside weather conditions, due to the tremendous amounts of fresh air required.

To change this plant over to Bunker C fuel oil, we installed duplicate oil pumping, heating and straining sets and provided steam coils in each of the three oil tanks. By using a very large heat exchanger, we are able to keep the oil at 210 F with steam at 10 lb pressure. The oil circulating system from the pump set is run to the

Typical piping layout for supplying heavy oil to process furnaces

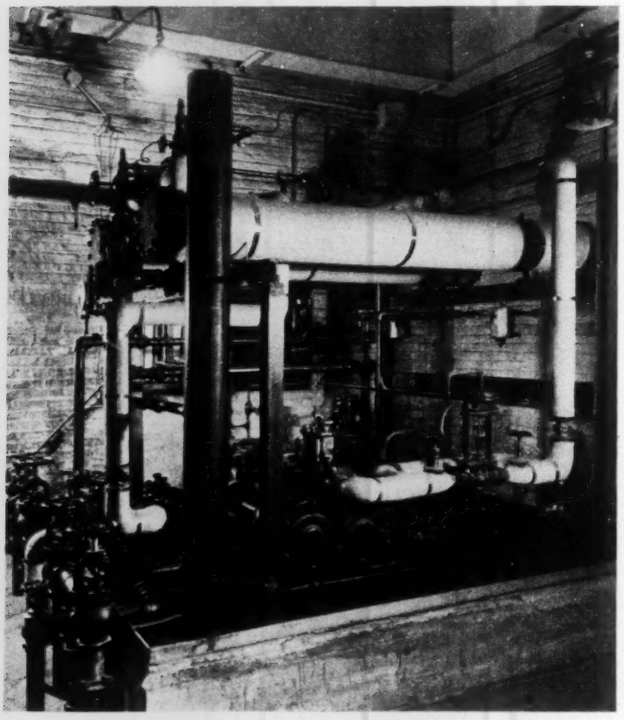


three boilers, which are in the same room. The main circulating line is brought down to each burner and then back overhead to leave clearance between the boilers. The line then goes into the kiln room and runs directly under each burner and overhead wherever passage is necessary. At a point halfway down the line a 5 kw electric heater is used to boost the oil temperature.

At the end of the circulating line, two pressure relief valves are inserted, cross-connected so that either may be used and the oil then runs into the oil return line and is taken back to the oil tanks. The oil discharge line which was used originally as an overhead circulating line was retained and the piping is cross-connected so that in the event of difficulty with the new circulating lines, the old lines may be used for emergency operation.

The entire line is covered with magnesia. Due to the length of the oil line, expansion joints are provided to prevent leakage or buckling of the pipe line. To further prevent temperature or pressure drop in this long line, sweep fittings are used throughout the entire system. The average oil consumption of this plant per furnace is approximately 8 gph and the maximum oil consumption in the boiler room is 100 gph, which means a full maximum oil consumption of about 244 gph. The oil pumping, heating, and straining set is designed to pump and heat 500 gph of oil. When the circulating line reaches the booster electric heater, 180 gal have been consumed, leaving 320 gal to be boosted with the electric heater. The oil temperature at this point drops to approximately 185 F. The electric heaters boost this quantity to the required temperature. Again, with this system, more successful operation is secured with Bunker C oil properly strained, heated, and pumped than had been previously secured with No. 5 oil under the old system.

Duplicate oil pumping, heating and straining set with one steam and one electric pump for circulating high pressure (300 lb) heavy oil. Note large standpipe for reducing pulsations in oil pressure when operating reciprocating pump



Piping Pitch for Boilers

The discussion here has been mainly on the use of Bunker C oils. There are plants, however, which are burning fuel oils (if they may be called oils) which are even heavier. The writer recently installed a system in the boiler room of a large forge plant which consisted of two 600 hp boilers operating at as high as 400 per cent of rating. This plant was originally installed to use Bunker C fuel oil under forced draft conditions. After the first six months of operation, the owners were advised that they could purchase pitch at a price advantage over Bunker C oil, and we were called in for advice as to the possibility of burning this fuel.

This pitch is perfectly solid at from 100 to 150 F and is so hard that it can only be cut with a hammer and a chisel. In making the study of the possibility of burning pitch we took the then present fuel and reduced the oil temperature as low as high efficiency could be maintained. At each temperature reading a viscosity test was made until we arrived at the maximum viscosity possible. We then took the pitch and heated it up until we approached the same viscosity as that of the fuel oil at its lowest efficient operating temperature. Viscosity readings were also taken at the oil temperature normally used in pumping the Bunker C oil from the tanks. The temperature of the pitch was then raised until the pitch attained the same viscosity. From this study, we learned that pitch may be pumped at a temperature of 300 F and may be burned efficiently at a minimum temperature of 365 F. The steam temperature available was 340 F, which obviously would not heat the pitch to the proper burning temperature but which could be used to keep the pitch from solidifying.

To change this plant over, we advised a separate, completely insulated oil storage tank with coils and oil suction heaters and the oil suction lines were completely steam jacketed. Duplicate oil strainers, also steam jacketed, were installed. A screw type, steam jacketed, fuel oil pump was selected for handling this material. The pitch is pumped through steam jacketed lines to a small oil tube boiler which is fired with a separate oil burner burning light fuel oil. The pitch is circulated directly through the tubes of this boiler and is thereby heated to 400 F. From this point, the pitch is circulated directly through steam jacketed control valves and steam jacketed lines to the oil burners and circulated from this point back to the storage tank. All steam jacketed hot oil lines are very heavily insulated.

This system for handling the pitch is entirely separate from that used for the Bunker C fuel oil. The Bunker C oil in this case is used as a light oil standby, and its lines are also brought directly to the burner through the customary oil pumping, heating, and straining equipment. In the event of any interruption in the supply of pitch or the operation of the pitch lines, Bunker C oil may be used immediately. The pitch is delivered to the plant in completely insulated tank cars which are heated to approximately 600 F and are so well insulated that they lose less than 100 F in 24 hr in zero weather. The plant burns an average of a car of pitch per day.

Study of Building Heating Economies Shows Money Value of Good Operation

By Earle Shultz* and J. C. Butler†

FOR the purpose of determining the extent to which one can make economies in the use of steam for heating buildings and of determining which of the many methods of accomplishing these economies gives the best results, a joint study was undertaken by the National Association of Building Owners and Managers and the National District Heating Association. To secure the necessary information, a questionnaire was prepared and distributed by the two associations.

Sufficient replies were received to furnish some general conclusions of interest and to indicate that a more complete study should be extremely valuable. One hundred and eighty-nine questionnaires were returned, of which 153 were sufficiently complete to be usable. Those that could be used were divided into two groups; 79 contained information as to steam consumption, both before and after the application of methods of heat control, while 74 gave only current data.

This report covers a study of the steam used for space heating only. Steam used for cooking, hot water and other non-space heating uses has been eliminated. As a number of replies did not separate heating steam from total steam, it was necessary for the committee to estimate this distribution. Where the consumption for the three summer months was given, this division has probably been made with reasonable accuracy. The necessity of estimating these figures, however, introduces some element of uncertainty in the results.

Analyses of the data by cities, by source of steam supply, by type of occupancy and by type of heating system were given in the October HPAC. The rest of the report is given this month.

Analysis by Type of Control

In the study shown in Table 1, the reports are grouped by type of control. It will be noted that all of the types of control except classes 6, 7 and 9 achieved results close to the average.

A comparison is also made between the controls using an intermittent or cycling type of operation, comprising classes 1, 2, 5 and 6, and those using a continuous flow or modulating control, comprising classes 3 and 4. The intermittent controls used 0.568 lb of steam, whereas the modulating controls used 0.525 lb. However, we cannot conclude from this analysis that the intermittent controls are less desirable than the modulating. In the first place, most of the modulating controls are considerably more expensive than the intermittent type, so that the net result, taking fixed charges into consideration, is probably about the same. In the second place, the modulating controls require the use of orifices (which in themselves show a saving of 6 per cent), and also a heating system well balanced and in good repair; whereas in most cases intermittent controls do not require orifices and the heating system is not overhauled when they are installed. In the third place, intermittent controls can be applied, at a reasonable cost, to the smaller buildings (that fundamentally use more steam per cubic foot of volume than the larger buildings) and to types of heating systems—such as one pipe, air line and gravity systems—which are the least efficient. Of the buildings reporting the use of modulating control, 27 were installed on two pipe vacuum, orifice vacuum and orifice gravity heating systems.

A total of the intermittent and modulating systems is also shown under the heading "Total of Controls Regulating Steam Supply," in Table 1. These systems used 0.555 lb of steam as compared with individual radiator control (class 7), which used 0.794 lb of steam. Control apparatus on the steam supply, therefore, shows a saving of some 30 per cent as against room thermostats or other types of individual radiator control.

One of the outstanding items of Table 1 is the result

To determine the extent to which economies in the use of steam for heating buildings can be made, a joint study was recently made by the National Association of Building Owners and Managers and the National District Heating Association. In brief, the following important conclusions may be drawn from it: (1) Careful and intelligent supervision of the use of steam is the first requisite in securing economies. (2) Steam meters should be installed on the supply to the heating system and daily pounds of steam per degree day records kept and compared with the standard. (3) The heating system should be maintained in good repair and if necessary overhauled so as to permit rapid circulation of steam at low pressures. This may involve the use of accurate and sensitive reducing valves and the installation of orifices in risers and radiators. All building air leaks should be eliminated. (4) A heating schedule, carefully worked out for each individual building, should be used. (5) Automatic heat control devices will definitely assure savings that will pay their cost in from two to four years

*Vice-president, Illinois Maintenance Co., and Chairman, Joint Committee on Heating Economies of the National Association of Building Owners and Managers and the National District Heating Association.
†Chief Operating and Commercial Engineer, Illinois Maintenance Co.

Table 1—Reports Grouped by Type of Control

Class of Control (See below)	No. of Bldgs.	Volume 1000 Cu Ft	Lb Steam per Deg Day		Lb Steam per Deg Day per 1000 Cu Ft		Per Cent Saving	Reports
			Before	After	Before	After		
1	13	37,473	26,506	22,180	0.707	0.592	16.3	Complete
	5	9,669		6,665		0.689		Partial
	18	47,142		28,845		0.612		Total
2	3	28,998	19,550	11,850	0.674	0.409	39.4	Complete
	1	3,692		2,410		0.653		Partial
	4	32,690		14,260		0.436		Total
3	4	14,816	12,275	7,195	0.828	0.485	41.4	Complete
	10	30,132		15,143		0.503		Partial
	14	44,948		22,338		0.497		Total
4	11	49,275	33,488	25,341	0.680	0.514	24.4	Complete
	4	6,159		4,905		0.796		Partial
	15	55,434		30,246		0.546		Total
5	18	73,783	59,426	45,062	0.805	0.611	24.1	Complete
	8	42,571		19,330		0.454		Partial
	26	116,354		64,392		0.553		Total
6	5	518	916	752	1.768	1.452	17.9	Complete
	7	17,444		13,670		0.784		Partial
	12	17,962		14,422		0.803		Total
7	5	24,139	23,375	16,800	0.968	0.696	28.1	Complete
	8	46,068		38,950		0.845		Partial
	13	70,207		55,750		0.794		Total
8	17	25,691	21,082	17,301	0.821	0.673	18.0	Complete
	26	45,811		25,593		0.559		Partial
	43	71,502		42,894		0.600		Total
9	3	7,196	5,650	5,440	0.785	0.756	3.7	Complete
	5	6,700		5,880		0.878		Partial
	8	13,896		11,320		0.816		Total
Intermittent Controls	39	140,772	106,398	79,844	0.756	0.567	25	Complete
	21	73,376		42,075		0.573		Partial
	60	214,148		121,919		0.568		Total
Modulated Controls	14	64,091	45,763	32,536	0.715	0.508	28.9	Complete
	15	36,291		20,048		0.552		Partial
	29	100,382		52,584		0.525		Total
Total of Controls Regulating Steam Supply	53	204,863	152,161	112,380	0.744	0.549	26.2	Complete
	36	109,667		62,123		0.566		Partial
	89	314,530		174,503		0.555		Total

Class of Controls—1, definite cycles automatically selected by outside temperatures. 2, definite cycles manually selected. 3, high vacuum control. 4, continuous flow pressure differential control. 5, radiator and outside uncontrolled cycles. 6, inside key thermostat control. 7, individual radiator control (thermostatic). 8, hand control of radiators or systems. 9, miscellaneous.

secured in class 8, where the regulation is entirely manual. These buildings use only 0.600 lb of steam. Most of these buildings operated on a planned schedule and had put their heating systems in good repair, in many cases installing orifices, pressure reducing valves and additional or more powerful vacuum pumps. Very careful supervision was then given to the proper adjustment of the steam pressures and vacuum in correlation with outside temperatures. This demonstrates that with a well balanced heating system, a competent engineer can, by carefully planned and controlled operation, secure results in line with that obtained by control devices. Contrasted with this, as will be brought out in more detail later in the report, many buildings expect control devices to produce results without supervision and with a faulty heating system.

Table 2—Effect of Building Size

No. of Buildings	Avg. Volume 1000 Cu Ft	Total Steam per Degree Day per 1000 Cu Ft
50	505	0.821
33	1,438	0.812
29	2,629	0.762
31	5,692	0.586
10	14,475	0.445

Table 3—Effect of Hours of Heating

No. of Buildings	Hours of Heating	Avg. Volume 1000 Cu Ft	Total Steam per Degree Day per 1000 Cu Ft
77	8 to 10	1,535	0.698
47	11 to 16	5,800	0.535
29	17 to 24	2,575	0.715

Size of Building and Heating Hours

Studies have been made of the effect of size of building and the number of hours of heating upon steam consumption. The effect of size is shown in Table 2. It is apparent that the larger buildings used a considerably smaller unit amount of steam than the small buildings, the largest buildings in the above table using only 54 per cent as much steam as the small ones.

A most interesting and rather surprising situation was brought out in the study of the effect of hours of heating on steam consumption, as shown in Table 3. From these figures it would appear that, as would be expected, there is a very definite saving in heating a building from 11 to 16 hours as against 17 to 24 hours. The unexpected thing is that a still further reduction in hours does not increase the saving, but, on the other hand, requires almost as much steam as long-hour heating. These figures would be more conclusive if they included only one type of building, such as office buildings, because it is probable that the inclusion of hotels in the long-hour group and of department stores in the short-hour group may distort the conclusions. The predominantly small buildings in the short-hour group as against the medium-hour group also influences the figures. However, it would seem that there is a limit to the saving that can be accomplished through reduction in hours of heating.

Cost of Control

Table 4 shows the relation of the cost of control apparatus to the size and steam consumption of a building. These figures include both the cost of changes in the heating system and of the control apparatus. They do not include those buildings where the only change was improvements in the heating system. This cost amounts to $0.98 per 1000 cu ft of building volume or $0.299 per 1000 lb of steam used, after the heating economies have been realized. The cost equals $0.75 per 1000 lb of steam saved.

Table 4—Cost of Control

No. of Bldgs.	Volume 1000 Cu Ft	Lb Steam per Deg Day		Lb Steam per Deg Day per 1000 Cu Ft		Per Cent Saving	Cost of Control	Reports
		Before	After	Before	After			
42	142,993	140,671	80,246	0.983	0.562	42.8	$126,941	Complete
18	30,014		23,295		0.776		42,885	Partial
60	173,007		103,541		0.598		$169,826	Total

Cost per 1000 cu ft volume	$0.98
Cost per 1000 lb steam (before)	0.214
Cost per 1000 lb steam (after)	0.299
Cost per 1000 lb steam saved	0.75

This figure does not necessarily hold true for the individual buildings. A few buildings indicated that with the medium-priced control the cost was recovered in one to two years, and with the higher priced and more complete systems the cost was recovered in from two to four years.

Heat Saving Without Control Apparatus

A summary of the information covering methods of heat saving without the use of control apparatus is as follows:

Most buildings made changes designed to eliminate air leakage from the building, such as caulking and weatherstripping windows, and reducing air leakage from the roof, stairways, elevator shafts and penthouses; and installed storm doors and storm windows in elevator shafts and other openings that could be closed. Some attention was also paid to control of heat required for ventilation.

Heating system changes were principally improvements in the circulation of steam. This was accomplished by stopping all leaks, eliminating pockets, cleaning and repairing the supply and return lines, and covering exposed piping. It also involved adjusting the size of the radiators in the various rooms to meet the heating requirements of the space. Many buildings installed orifices in the radiators and in their distribution lines, and installed special supply lines to first and second floors and other portions of the building requiring long-hour heating. Some zoned the building into two or more sections, while a few changed over one pipe systems to two pipe systems. They also installed better and more sensitive reducing valves on the steam supply and put in new and additional vacuum pumps to increase the vacuum and permit better control of the steam differential.

Table 5—Schedule of Hours of Heating (Composite of All Reporting)

Daily Temperature, F	Week Day		Sunday	
	On	Off	On	Off
0	2:30 a.m.	10:00 p.m.	3:30 a.m.	8:30 p.m.
20	4:20 a.m.	8:00 p.m.	6:00 a.m.	6:00 p.m.
40	6:10 a.m.	6:30 p.m.	7:40 a.m.	4:20 p.m.
50	7:00 a.m.	5:15 p.m.	9:00 a.m.	2:10 p.m.
60	8:30 a.m.	3:30 p.m.	10:00 a.m.	1:00 p.m.

Table 5 shows a composite schedule of hours of heating of those buildings reporting. A total of 90 or 58.8 per cent of the buildings used a heating schedule in some form. Some of the buildings not reporting one doubtless depend upon the control device to provide such a schedule. A heating schedule correctly worked out for each individual building is a fundamental necessity in securing heating economies.

Importance of Records and Meters

While many of the buildings reporting probably did not indicate the records they actually use, it is apparent that less than half of the buildings keep records from which heating efficiency can be determined, but depend wholly on over-all cost figures. Records of steam per degree day and the use of a chart by which actual consumption can be plotted against theoretical requirements are used by only 30 per cent of the buildings. The reason for this probably lies in the fact that many buildings do not have steam meters. This applies particularly to those buildings operating their own plants where the only record is of coal consumption. From the reports it is apparent that many such buildings do not distinguish between economies in the production of steam in their boilers as against economies in the utilization of steam in their heating systems. In some cases where a reduction in heating costs was claimed, it was found that the building was actually using more steam per degree day than before but that the increased cost had been offset by some improvement in boiler efficiency.

Inasmuch as the amount of saving to be made is affected much more by the quantity of steam used than by its efficiency of production, all buildings should be equipped with steam meters so that they can determine the actual amount of steam used for heating. This cannot be done accurately from the coal records, because steam quantities determined from coal records are usually based upon assuming a certain rate of evaporation, which may or may not be correct.

The most important information for the control of steam utilization is the daily quantity of steam used for heating only. This figure, divided by the degree days for the day, gives the building's steam consumption per degree day. This result can be compared with a standard obtained by multiplying by 0.6 the volume of the building in 1000 cu ft.

Performance of Controls

The answers to questions contained in the third part of the questionnaire applying to the "Control Apparatus" are summarized as follows:

All types of apparatus were reported upon favorably. With most of them, however, there were cases of mechanical difficulties which were later adjusted. In many cases of criticism of the results secured by any particular device, it is possible that failure to obtain good results was due to a lack of supervision, improper operation of the device, or to faults in the heating system itself. In the installation of many of the controls two difficulties arose. The first was in securing a proper balanced flow of steam through the heating system. This was not the fault of the device and was corrected by a careful overhauling of the heating system. The second was in securing a proper calibration or adjustment of the control. This was accomplished by trial and the making of the necessary adjustments to produce the best results.

While many buildings use the intermittent control with splendid results, the "off" period of their cycle presents some difficulties. If the cycle is too long and the radiators become cold for a considerable period of time, tenants opening their windows during this off period may have their offices cooled. Some tenants complained of this situation, and in some cases it took a year to educate them to keep their windows closed. Unless the heating system is adjusted for rapid steam circulation, if the cycle is short, the interval during which the steam is on may not be sufficient to force steam to the far ends of the system, with the result that one part of the building may be overheated while another part is underheated. This was to some extent overcome by orificing the risers and radiators. These difficulties in the use of intermittent control do not occur with those controls using a continuous flow of steam. It further emphasizes the

necessity of keeping the heating system itself in good repair.

Another source of trouble arises in securing a large enough steam flow during the heating up period in the morning. Some buildings reported that it is necessary to operate the system manually during this period. Another comment was that some of the more delicate systems, designed to control the steam pressure to fractions of an ounce, have to be reset in mild weather in order to prevent overheating and underheating.

General Conclusions of Study

Under the fourth part of the questionnaire, the general conclusions may be stated briefly as follows:

While money saving is one of the important benefits of heat control, even more emphasis was laid upon the improvement in service and comfort. Heat control produces a steady, even heat without periods of overheating and underheating, which is more healthful and stimulating and very much appreciated by the tenants. Some labor saving is also accomplished through the use of controls, by eliminating night engineers to turn steam on and off.

Nearly all buildings agreed that efficient supervision of the heating system is most important if economies are to be made. The control apparatus ranked next, with good records and changes in the heating system third.

In reply to the question as to the possible benefits of purchasing steam, almost all of those buildings purchasing steam pointed out values. In one or two cases it was reported that the cost of purchased steam was too high and plants had been installed. Those buildings operating their own plants either made no comment on the values of purchased steam or felt that private plant operation was more efficient. Some of the values of purchasing steam enumerated were:

1) Better service because of the constant availability of steam.

2) The securing of greater economies in heat utilization because of the elimination of plant wastes and the ability to secure closer control.

3) Securing greater efficiency, through the cooperation of the local heating company in educating the building engineers was pointed out as being very helpful. Particularly in mild weather, economies can be more easily realized.

4) The purchase of steam also results in the elimination of smoke; coal and ash handling; saves on labor, repairs, insurance and inspection; avoids possible shutdowns or labor troubles; and requires less supervision and worry.

5) Buildings are cleaner, with lower decorating costs and less damage to merchandise in the buildings.

6) Many buildings also secure rental from space which would otherwise be occupied by the boiler plants and stack.

A brief summary of the important conclusions that may be drawn from the study of heating economies are:

1) Careful and intelligent supervision of the use of steam is the first requisite in securing economies.

2) Steam meters should be installed upon the supply to the heating system and daily pounds of steam per degree day records kept and compared with a standard.

3) The heating system should be maintained in good repair and, if necessary, overhauled so as to permit rapid circulation of steam at low pressures. This may involve the use of accurate and sensitive reducing valves and the installation of orifices in the risers and radiators. All building air leaks should be eliminated.

4) A heating schedule carefully worked out for each individual building, should be used.

5) Automatic heat control devices will definitely assure savings that will pay their cost in from two to four years.

In conclusion the committee recommends that this study be continued for another year in the hope of securing a much larger number of replies. With an adequate number of reports, it should be possible to determine rather conclusively the comparative efficiency of the several types of heating systems and control devices. This study could be made by local committees, who could discuss the statistics turned in for local buildings with their engineers and in that way interpret the figures and conclusions much more accurately than could a national committee.

Edison Electric Institute Reports Number and Horsepower of Air Conditioning Installations

Following is a summary of the total air conditioning installations reported by 141 electric utility companies to the Edison Electric Institute as of March 31, 1939. The complete report, which is noted in our Book Reviews column this month, also gives the installations prior to 1938 and during 1938 according to the territories of the various utility companies.

Application	Total as of 3/31/39 No.	Hp*
Residential		
Private Homes	16851	17210.80
Commercial		
Apartments	60	1727.25
Banks	373	18247.01
Barber & Beauty Shops	465	3408.97
Broadcasting Studios	122	4404.93
Clubs	207	4401.29
Funeral Homes	509	5159.05
Hospitals	314	4611.34
Hotels	802	40397.80
Office Buildings	657	86293.52
Offices, Doctors & Dentists	644	1690.05
Offices, Miscellaneous	5462	44695.67
Public Buildings	308	47781.33
Recreational	153	2860.30
Restaurants	3253	61879.27
Stores, Department	660	76405.24
Stores, Drug	621	9185.19
Stores, Retail Misc.	3630	64692.91
Theaters	1648	151243.47
Other Commercial	1246	24632.74
Industrial		
Candy Mfg.	157	9413.67
Drug & Chemical Mfg.	49	2329.30
Printing & Lithographing	92	5846.46
Textile Mfg.	38	1839.25
Tobacco Mfg.	27	1495.43
Metal Working	3	16.00
Food Processing	78	2782.35
Fur Storage	86	525.01
Other Industrial	1232	65674.88
Total Installations	39747	760851.38
Installations included above that use power other than central station electric	21	4773.00
Self-contained conditioners included above	8652	18226.71

*Hp includes all auxiliaries.

The Psychrometric Chart

Its Application and Theory

By William Goodman*

HERETOFORE, only the area to the right of the saturation curve has been regarded as the useful portion of the psychrometric chart; the area to the left has generally been ignored as having no useful application.

Nevertheless, the area to the left of the chart plays a significant part because it represents the condition of fogged air. To the right of the saturation curve, any point on the chart represents the state of mixtures of air and superheated steam—that is, clear air. On the saturation curve itself, any point represents a mixture of air and saturated steam. To the left of the saturation curve, any point represents a mixture of air, saturated steam, and free moisture held in suspension in the air as fine liquid particles—in short, fog.

Fogged Air

In the fogged state the air is saturated and, in addition, is in intimate contact with finely divided drops of water. Inasmuch as the air and water are intimately mixed, the temperature of the water must be the same as the temperature of the saturated air.

The temperature of fogged air is read along the lines of constant temperature of adiabatic saturation. The lines of dry bulb temperature cannot be extended into the fogged field of the chart. One way of visualizing the fact that the temperature of fogged air must be read along the lines of constant temperature of adiabatic saturation—or, as they are frequently called, lines of constant wet bulb temperature—is to consider the case in which liquid is sprayed into unsaturated air. Assume that the water spray consists of an extremely fine mist. If the temperature of the liquid is the same as the wet bulb temperature of the air, the final state of the air will be represented on the psychrometric chart by a point on a line of constant wet bulb temperature. The dry bulb temperature of the air will fall as it surrenders the heat necessary to evaporate the liquid into the air. If just enough water is supplied to saturate the air, the dry bulb, wet bulb, and dew point temperatures of the air will all be equal to each other. If more water is supplied than is required to saturate the air, the excess water will not vaporize; it will be entrained. Therefore, the temperature of the already saturated air will not change as it entrains water which is at a temperature equal to the temperature of the saturated air itself.

Now in Fig. 14 point *1* represents the condition of just-saturated air, and point *2* the condition of fogged air. Both points lie on the ratio line *1-2*. Inasmuch as the air is saturated in both states, and the entrained water is at the same temperature as the saturated air, the slope of the ratio line *1-2* in the fogged field must be determined by equation 14. From the discussion given under "Wet Bulb Temperature," it is apparent that for air in any condition, there is only one ratio line for which the water temperature is equal to the temperature of the saturated air and that this ratio line corresponds to a line of constant temperature of adiabatic saturation, or so-called wet bulb temperature. Hence, the temperature of fogged air is constant along any one line of constant wet bulb temperature.

In order to determine the temperature of a point in the fogged field of the chart, the two wet bulb lines located nearest to it must be drawn. For example, to find the temperature of fogged air in the state represented by point *1* of Fig. 15, the wet bulb lines corresponding to points *2* and *3* must be drawn. The temperature of the fogged air can then be read. Because the lines of wet bulb temperature are very nearly parallel to the lines of constant enthalpy,

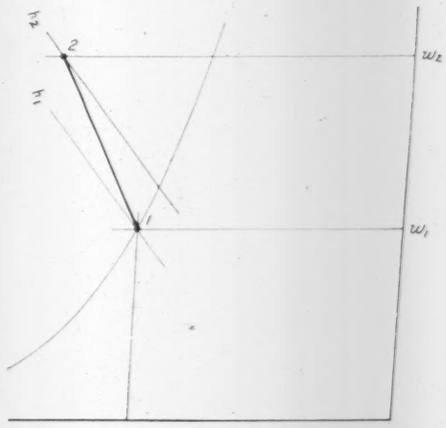


Fig. 14—Illustration showing method of reading temperature of saturated air in fogged field of chart along lines of constant wet bulb temperature†

†See table of symbols, p. 358, June issue.

In addition to the elementary uses of the psychrometric chart—such as showing the relationships between dry bulb, wet bulb, and dew point temperatures, and relative humidity—it provides a simple and enlightening means of analyzing many complex problems in comfort and process air conditioning, including drying. . . . New psychrometric charts covering the low, middle and high temperature ranges, with saturation curves for various barometric pressures, are presented here. The general theory of the psychrometric chart is discussed in detail, and the methods of solving problems by the use of charts are described. . . . These new charts facilitate solution of most problems and lead to clearer understanding of the fundamentals

*The Trane Co. Member of Board of Consulting and Contributing Editors.

Part 6. Part 1 was published in HEATING, PIPING AND AIR CONDITIONING, June, 1939, pp. 357-360; Part 2 in July, pp. 421-424; Part 3 in August, pp. 485-487; Part 4 in September, pp. 549-551; and Part 5 in October, pp. 613-615 and p. 617.

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the temperature of the fogged air can frequently be read closely enough without even drawing the lines of wet bulb temperature—just as in the case of unsaturated air.

Regardless of whether the air is unsaturated or fogged, the weight of moisture mixed with the air is always read along the horizontal line of moisture content which runs through the point in question. Thus, for the fogged air represented by point *1* of Fig. 16, the total weight of moisture mixed with air is read along the horizontal line through point *1*. This total weight of moisture consists of both free liquid moisture suspended in the air and saturated vapor mixed with the air. On the other hand, the weight of saturated vapor mixed with the air

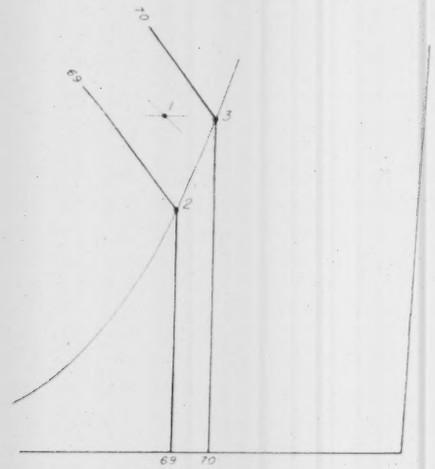


Fig. 15—Method of determining the temperature of saturated air whose state is represented by a point in the fogged field of the chart

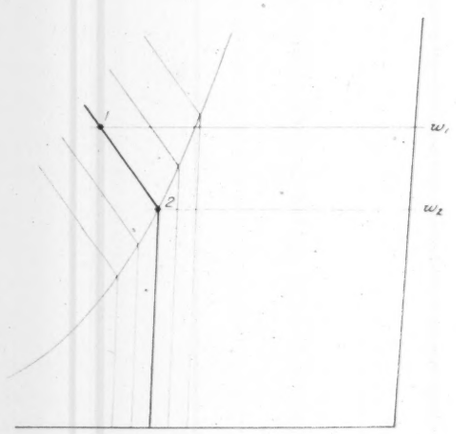


Fig. 16—Method of determining amount of saturated vapor and amount of free moisture entrained in the form of liquid in fogged air

is always read at the point where the line of wet bulb temperature intersects the saturation curve. For air in the condition represented by point *1* of Fig. 16, the weight of saturated vapor mixed with the air is read at point *2*, where the line of constant wet bulb temperature through point *1* intersects the saturation curve. The reason for this is that at points *1* and *2* the temperature of the saturated air is the same. Hence, the maximum weight of saturated vapor that can be mixed with the air is the same at both points. The weight of free moisture suspended in the air in liquid form is given by the difference between the absolute humidities at points *1* and *2* —that is, the weight of liquid suspended in the air is $(w_1 - w_2)$. The weight of saturated vapor mixed with the air is w_2.

Equation 12 in the section "Mixtures of Air and Water" is equally valid if the liquid is evaporated into the air or if it is merely entrained—that is, carried along by the air in the form of a fine mist. If water at a temperature which differs from the temperature of the saturated air is entrained by the air, the final condition of the air will lie along a ratio line, the slope of which is determined, as heretofore, only by the initial enthalpy of the water. Problems involving both fogged and clear air are solved in exactly the same manner as in the preceding examples. The only difference will be that the ratio lines instead of lying wholly in the clear field of the chart will extend into the fogged field.

In Fig 17 point *1* represents the initial condition of unsaturated air. If the air is passed through a spray of moisture whose temperature differs from the wet bulb temperature of the air, the final condition of the air leaving the apparatus will lie along the ratio line *1-3*. The slope of this ratio line is determined only by the initial enthalpy of the moisture as heretofore. If insufficient water to saturate the air is supplied, the final state of the air leaving the apparatus will be represented by, say, point *2* of Fig. 17. If just sufficient water is supplied to saturate the air, the final state of the air will be point *3* on the saturation curve. On the other hand, if more water is supplied than is needed to saturate the air—and it is broken into a fine mist so that it is readily entrained by the air—the final state of the air will be point *5*. The important thing to note is that points *2*, *3*, and *5* all lie on the same ratio line.

The dry bulb temperature of the unsaturated air initially in state *1* will fall as moisture is added because the air surrenders the heat needed to evaporate this moisture. Thus, the dry bulb temperatures at points *2* and *3* are lower than at point *1*. If the initial temperature of the water is higher than the wet bulb tempera-

Fig. 17—Ratio line extended into fogged field when moisture is atomized in air. Angles between lines and distances between lines are exaggerated for the sake of clarity

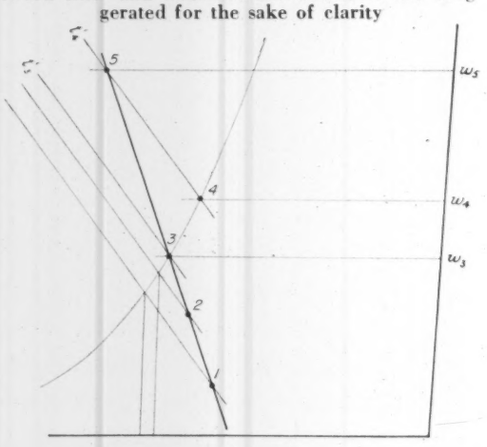
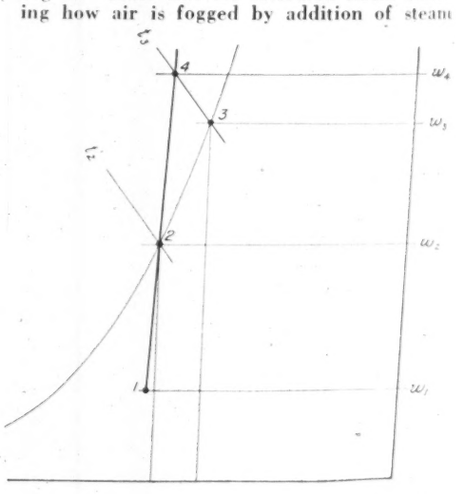


Fig. 18—Ratio line for saturated steam showing how air is fogged by addition of steam

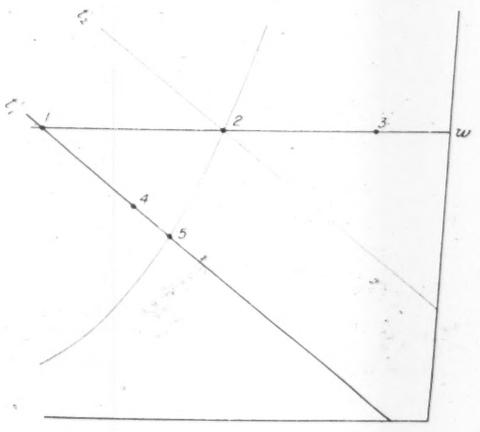


ture of the air in state *1*, then the wet bulb temperatures as well as the enthalpies at points *2* and *3* must be higher than at *1* because of the heat added by the moisture. From points *3* to *5* the water only mixes with the air; there is no evaporation of moisture into the saturated air. Nevertheless, the temperature of the saturated air must increase if the temperature of the water is higher than the initial wet bulb temperature of the air. Consequently, the temperature of the saturated air as read on the wet bulb temperature line *4-5* through point *5* is higher than the wet bulb temperature line through point *3*. Inasmuch as the temperature of the saturated air increases, some further moisture must be evaporated if the air is to remain saturated—as it must. For air in the state represented by point *5*, the quantity of saturated vapor mixed with the air is read at point *4*. Point *4* is located where the wet bulb line through *5* intersects the saturation curve. The amount of liquid moisture in suspension in the air is the difference in the absolute humidities at points *5* and *4*, that is the quantity $(w_5 - w_4)$.

As previously shown, equation 12 is equally applicable whether the moisture is added in the form of water or steam. If more steam is added to a given weight of air than is required to saturate the air, some of the steam will condense and fogged air will result. The water formed by the condensation of the steam will remain in the form of fine drops suspended in the air. The change in the state of the air for the case in which steam is sprayed into the air is illustrated in Fig. 18. Point *1* represents the initial state of the air. The enthalpy of the steam is such that the ratio line is *1-4*. As steam is added to the air, the point representing the state of the air moves up along line *1-4* until it passes the saturation curve at point *2*. If additional steam is added to the air, the final condition of the air leaving the apparatus will be represented by point *4*.

As long as the air remains unsaturated, the steam will mix with the air and there will be no condensation and little change in the dry bulb temperature of the air. However, once the air becomes saturated at point *2*, the additional steam that is added to the air must condense. In other words, as the state point of the air moves up from *2* to *4*, some of the steam must condense. As a result of the condensation of part of the steam, latent heat is liberated, and this increases the temperature of the saturated air. The temperature of the saturated air in the condition represented by point *4* is read along the line of wet bulb temperature *3-4* through point *4*.

Fig. 19—Illustration showing path of state point when fogged air is cleared either by heating or by mechanical separation



The weight of liquid moisture suspended in the air is the difference between the absolute humidities at points *3* and *4*. Point *3* is located where the wet bulb line through *4* intersects the saturation curve. The weight of saturated vapor mixed with the air is the absolute humidity at point *3*.

The following example clearly shows that the total quantity of heat surrendered by the steam is equal to the heat gained by the air when the two are mixed.

Example 8: Steam is sprayed into air which is at an initial dry bulb temperature of 55 deg and a dew point temperature of 35 deg. The weight of steam added amounts to 25 lb per min. The weight of air flowing through the conditioner is 1875 lb per min. The barometric pressure is 29.92 in. The steam is saturated at a pressure of 26 lb gage. Find the final temperature of the air and show that the heat released by the steam is equal to the increase in the enthalpy of the air-vapor mixture.

Solution: For saturated steam at 26 lb gage, $h_s = 1170$ Btu. Referring to Table 1 for $q = 1170$, $\theta = 85°\text{-}39'$. Locate point *1* on the psychrometric chart of Fig. 18 at the initial condition of 55 deg dry bulb and 35 deg dew point temperature. Draw the ratio line *1-4*.

$$\text{Lb of steam added per lb of air} = \frac{25}{1875} = 0.01333 \text{ lb of steam}$$

For the initial dew point temperature of 35 deg, $w_1 = 0.00426$ lb.

Using equation 2, $w_2 = 0.00426 + 0.01333 = 0.01759$ lb.

Locate point *4* where the horizontal line for an absolute humidity of 0.01759 lb crosses the ratio line *1-4*.

Inasmuch as point *4* lies in the fogged field, the final temperature of the fogged air is read on the wet bulb line passing through point *4*. Thus, the final temperature of the fogged air is 69.2 deg as read on the psychrometric chart.

Although the total weight of vapor mixed with the air amounts to 0.01759 lb as read at point *4*, the weight of saturated vapor mixed with the air is less than this. The weight of saturated vapor mixed with the air is read at point *3* where the wet bulb temperature line of 69.2 deg intersects the saturation curve. The weight of saturated vapor as read at point *3* is 0.01534 lb.

The weight of liquid mixed with the air is $w_4 - w_3 = 0.01759 - 0.01534 = 0.00225$ lb per lb of dry air.

The weight of saturated vapor added to the air is $w_3 - w_1 = 0.01534 - 0.00426 = 0.01108$ lb per lb of dry air.

From the steam tables, the initial enthalpy of the steam added to the air is 1170 Btu, and the enthalpy of the saturated vapor at the final temperature of 69.2 deg is 1092 Btu.

Heat surrendered by vapor = 1170 — 1092 = 78 Btu per lb of steam added.

Final enthalpy of liquid at 69.2 deg is 37.2 Btu per lb of moisture.

Heat surrendered by condensing liquid = 1170 — 37.2 = 1133 Btu per lb of moisture.

Total heat surrendered by steam = $78 \times 0.01108 + 1133 \times 0.00225 = 3.41$ Btu per lb of dry air.

From the psychrometric tables of the article of Footnote 6, specific heat of humid air at 35 deg dew point = 0.2419. Heat gained by air and vapor initially mixed with it in heating from 55 deg at point *1* to 69.2 deg at point *4* is 0.2419 (69.2 — 55) = 3.43 Btu per lb of dry air.

From this it is evident that the heat surrendered by the steam

is equal to the heat gained by the air and the vapor initially mixed with it.

For wet steam (that is, a mixture of steam and water) the ratio line would of course lie somewhere between the line *1-5* of Fig. 17 and the line *1-4* of Fig. 18. The method of reading the temperatures, the weight of free liquid entrained in the air, and the weight of saturated vapor mixed with the air is exactly the same as described in the preceding paragraphs.

The problem of keeping rooms clear of fog or of clearing up fogged air frequently presents itself. This can be accomplished in one of three ways.

1) By heating the fogged air.
2) By separating the water from the fogged air mechanically.
3) By mixing the fogged air with warmer unsaturated air.

The first method is illustrated on the psychrometric chart of Fig. 19. If fogged air in the state represented by point *1* is heated, the moisture will evaporate and the resulting condition of the air will be represented by a point such as *3* on the horizontal line *1-3*. The point representing the condition of the air must move along the line *1-3* of constant absolute humidity because moisture is neither added to the air nor removed from it during the heating process. The temperature of the air will increase.

In the second case, moisture can be removed by means of mechanical separation only if the liquid suspended in the air is in the form of very coarse drops. Fine drops in suspension in the air, as is the case when air is fogged, probably cannot be separated from the air to any degree in this manner. Of course, only free moisture can be removed from the air by mechanical separation; saturated or superheated vapor mixed with the air cannot be removed in this way.

In mechanical separation the temperature of the air does not change as it passes through the liquid separating equipment. Hence, the final condition of the air leaving such a separator is represented by a point such as *4* along the line of constant wet bulb temperature *1-5* of Fig. 19. If all of the free liquid could be removed, the air would leave the separating apparatus in the saturated condition represented by point *5*.

The third method requires the mixing of warm, unsaturated air with the fogged air in order to evaporate the moisture. The result of mixing warm, unsaturated air with fogged air is discussed in detail in a later section.

[*To be continued*]

Diesel Engines to Heat Snow Cruiser

Although they will be stationed for three years in the coldest spot in the world, the four scientists who will work and live aboard the snow cruiser—the $150,000 survey and research unit of the U. S. government's forthcoming Antarctic expedition—are not worrying about keeping warm.

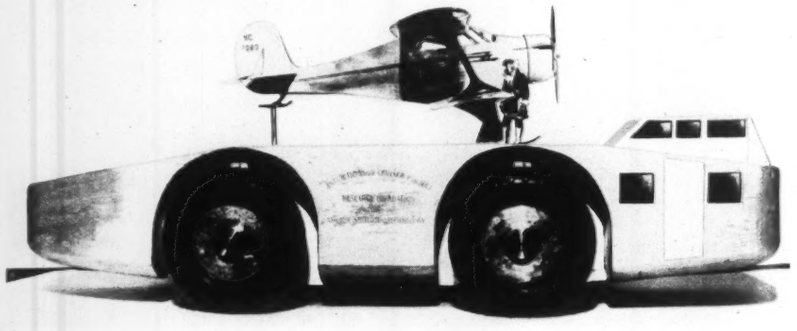
The staff of the research foundation of Armour Institute of Technology, designers of the cruiser, has made adequate provision for heating. Both of the two 150 hp diesel engines which, directly connected to traction-type generators, furnish motive and auxiliary power, have been equipped with a radiator cooling system. The antifreeze solution, heated by circulation through the engines, will be piped to radiators in the various compartments of the cruiser, assuring comfortable room temperatures at all times.

Dr. Thomas C. Poulter, scientific director of the research foundation and second in command of the last Byrd Antarctic expedition, will command the cruiser and he will be accompanied by three other scientists. They will study various natural phenomena and map and explore vast areas of Antarctica for the government.

The cruiser is 55 ft long and loaded, it will weigh approximately 180,000 lb. The cruiser has storage space for a year's food supply, 2500 gal of diesel fuel oil and 1000 gal of gasoline, living quarters, galley, radio station, darkroom, engine room, machine shop and scientific laboratory.

The two diesel-electric units furnish power to the traction motor on each of the 10 ft rubber tired wheels in any combination of one, two, three or four wheels. By means of an ingenious arrangement of hydraulic lifts, the wheels are retractable and the cruiser will be able to span 15 ft crevasses in the ice. The steering mechanism consists of two levers, one controlling the rear wheels and the other the front wheels. The cruiser is designed to be able to turn around in its own length, move laterally at an angle and climb 37 per cent grades. It is designed to make between 10 and 30 mph and have a cruising range of 5000 miles.

Moored on the top deck will be a five passenger plane, fully equipped with cameras and scientific instruments. The plane will operate as far as 600 miles from its mobile base, making it possible for the crew to increase greatly the scope of their explorations.



The snow cruiser is 55 ft long and when loaded will weigh approximately 180,000 lb. It contains storage space for a year's food supply, diesel fuel oil and gasoline stores, living quarters, galley, radio station, darkroom, engine room, machine shop, and a scientific laboratory. Its airplane can operate 600 miles from its mobile base, making it possible to increase greatly the scope of the Antarctic explorations

Corrosion in Steam Heating Systems

Dissolved Gases—The Basic Causes for Corrosion

By Leo F. Collins* with Everette L. Henderson†

THAT the theory of heating system corrosion propounded last month is substantially correct has been demonstrated both by experimental data and by those obtained from operating systems.

Experimental Data—From a practical point of view the most important support of the theory is the observation that a CO_2 laden condensate moving along in a piping system eventually dissolves enough iron to become inactive. To determine experimentally that prolonged contact with iron surfaces raises the pH value of a condensate and simultaneously reduces its corrosive inclinations, the apparatus of Fig. 4 was supplemented by the addition of four iron loops just preceding the trap. A corrosion specimen was placed in each loop and analyses of the condensate were made. The data of Fig. 7 show the results of one such test. This is convincing evidence in support of the theory.

Data from Operating Systems—In general the data from all locations agree favorably with the theory. But before these are considered, it will be more instructive to review in detail the conditions encountered at two particular locations:

1) At location No. 14 a short steel nipple and the attached elbow in the condensate line immediately adjacent to a water heater fails about every two years. The attached piping is always clean and very evenly thinned along those surfaces where the condensate first makes contact. The failures always occur at the threaded end of the nipple close to the heater. Little or no evidence of corrosion is manifested in this line at a distance of 10 ft or more from the heater. A steam pressue of 5 lb ga is perpetually maintained, and condensation rates as high as 3000 lb per hr have been measured.

The first corrosion measurement made at this location under normal operating conditions showed a rate of 39.2. The specimens when removed showed a bright metallic sheen. This is an evidence of acid attack with the evolution of hydrogen. Analyses of the condensate gave the average result shown for the zero venting rate in Table 2. No oxygen could be found with the Winkler method either in the steam or in the condensate. The CO_2 was determined by direct titration with a standard alkali and the specific conductance of the condensate was also measured. After subtracting the conductance due to water, the corrected values, due to carbon dioxide, were compared with the CO_2 conductance relationships proposed by Walker and Cormack[4] and by Kendall[5]. The agreement was such as to indicate that only CO_2 was present.

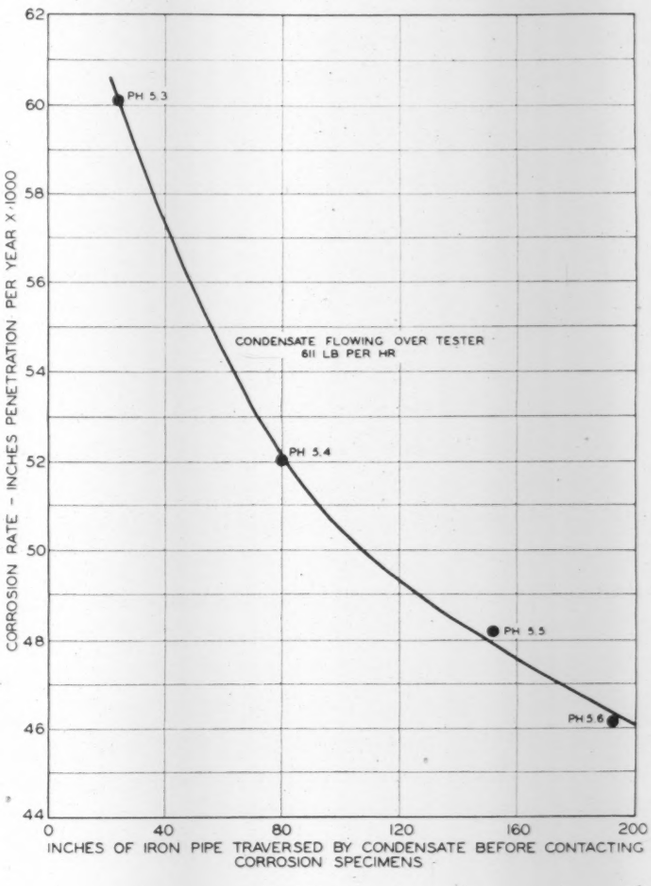
No provision was made for venting non-condensable gases from this heater. Thus, at high condensation rates considerable carbon dioxide accumulates in the vapor space. As a result more is dissolved in the condensate than would occur if a vent had been provided. As an experiment a vent was installed and corrosion rates were measured when different quantities of steam were bled from the vapor space. These results are also shown in Table 2.

It is significant that such low concentations of carbonic acid caused corrosion of the magnitude found and that when the pH value approximated that indicated by theory the corrosion rate decreased to almost a "mild" state. It is felt that the high operating temperature (190 to 225 F) and high condensation rates account for the small deviation from the limit indicated by theory. A vacuum heating system in this building is relatively free of corrosion troubles.

2) At location No. 17 the same steam is used as at location No. 14. A water heater which *operates intermittently* has experienced no corrosion failures in the return line or elsewhere, but in the two pipe gravity heating system which operates continually for long periods, a number of corrosion failures have occurred. All of these have been in the coils of unit heaters and in return lines close to radiators. It is estimated that about 50 per cent of the branch return lines failed within a period of 10 years, and failures of the copper alloy bellows of thermostatic traps have been even more frequent.

A corrosion measurement made in the bottom of a radiator

Fig. 7—Experimental data showing the increase in pH value and decrease in corrosiveness of freshly formed condensate as a function of progressive contact with iron surfaces



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Part 3. Part 1 was published in HEATING, PIPING AND AIR CONDITIONING, September, 1939, pp. 539-542; and Part 2 in October, pp. 620-622.

[4]J. Walker and W. Cormack, *Journal American Chemical Society*, Vol. 77, No. 2 (1900)

[5]J. Kendall, *Journal American Chemical Society*, Vol. 38, 1480 (1916).

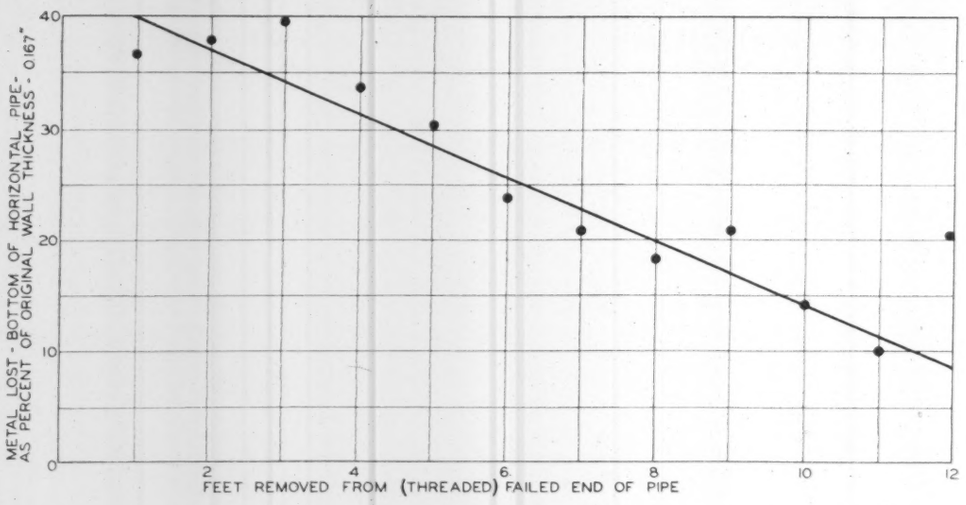


Fig. 8—Amount of metal dissolved from the bottom of a horizontal return line at varying distances from the condensing equipment—Building No. 17, pipe life about nine years

gave a value of 24.1, which represents an active corrosion condition and predicts that failures should occur. The run-out from this radiator did fail after eight years. Perhaps the most informative failure of all was that which occurred in a small horizontal return line handling condensate from a battery of radiators in this building. The failure was (as invariably happens) at the threaded end closer to the radiators. The bottom of the pipe was distinctly grooved and the groove appeared to be less pronounced at the tail end of the pipe. Following its removal, the pipe was sawed into 1 ft lengths and the thickness of the metal at the bottom of the groove was measured for each segment. The amount of metal lost from the bottom of the groove, as per cent of the original wall thickness, is plotted in Fig. 8 against the distance each measurement was made from the failed end. This plot plainly shows that corrosion diminished as the condensate moved down the pipe.

Table 2—Corrosion Rates and Average Analyses of Condensate[a] as a Function of the Amount of Steam Bled from the Vapor Space of the Water Heater in Building No. 14

Rate of Venting lb/hr	Oxygen cc/l	Carbon Dioxide ppm	pH Value	Corrosion Rate (In./Yr) x 1000
0	0.0	3.0	5.7	39.2
55	0.0	1.8	5.8	31.1
100	0.0	1.4	6.2	20.3

[a]Steam contained an average of 16 ppm of CO_2 and no oxygen.

In Fig. 9 all the corrosion rates measured in return lines are plotted against the pH values of the respective condensates*. The oxygen concentrations which persisted are also noted for each point. Unfortunately the amount of solution flowing in each case was not measured; for this reason the chart must be regarded as indicative of a relationship between pH and corrosion, rather than as a mathematical correlationship of these factors. Nevertheless it is believed that the following points are substantiated, all of which have been previously enunciated:

1) CO_2 does not cause "active" corrosion when the pH value of the condensate is above an approximate value of 5.9.

2) For pH values below the critical, CO_2 rather than oxygen is the controlling corrosive solute.

3) Because of operating conditions, the pH value of the condensate in *main* return lines of common types of heating systems is seldom below the critical value even though the CO_2 content of the steam supply may be high.†

With respect to item 3 it should be carefully noted that this does not hold true for "unusual" locations, or unusual types of systems nor for condensing apparatus and return lines immediately adjacent in many of the more common types of systems.

With respect to items 1 and 2 it should not be inferred that pH values above the critical necessarily presage freedom from corrosive attack. If oxygen is present corrosion will proceed as is attested by the data in Fig. 10. Here corrosion rates measured in return lines, where the pH values of the condensates were higher than 5.9, are plotted against dissolved oxygen concentrations. The inflection of this curve indicates the aggressiveness of oxygen per unit of gas becomes progressively less as the concentration increases. Nevertheless it is plain that oxygen may eventually cause failure if the velocity of flow and other conditions are such as to remove the products of corrosion which otherwise offer some protection.

Effects of System Design and Operation

Since corrosion varies in different parts of the same system and in different types of systems, it may be correctly assumed that the amount of gas dissolved in the condensates also varies. The amount dissolved is governed largely by the design and methods of operation of the heating system; that is, whether it is continuous or intermittent and whether pressure or vacuum is employed. Because of such variations and the many possible combinations thereof which are influential, it is

*Analyses were not made at all locations, hence all the corrosion rates shown in Fig. 3 could not be plotted in Fig. 9.

†The steams used in the systems investigated varied from as little as 1 ppm to as high as 25 ppm of CO_2.

This comprehensive treatment of the subject of corrosion in steam heating systems is based upon about 15 years of research work for The Detroit Edison Co. and is believed to be the most authoritative treatise on the subject that has ever been prepared. . . . Engineers concerned with operation and maintenance of building heating systems, district heating engineers, and industrial piping engineers, will find the information as useful as will designers and installers of heating plants. . . . Because of its length it is necessary to publish this treatise in serial form. Accordingly, the reader is cautioned not to pass final judgment on any statement or data until the publication of the series is completed

impossible to correlate dissolved gas with different types of systems. The best that can be done is to cite general observations.

In main return lines the amount of oxygen dissolved by the condensate is usually several times more than can be accounted for upon the basis of that entrained with the steam. Particularly is this true for vacuum systems and those which operate intermittently. In both cases infiltration of air is responsible.

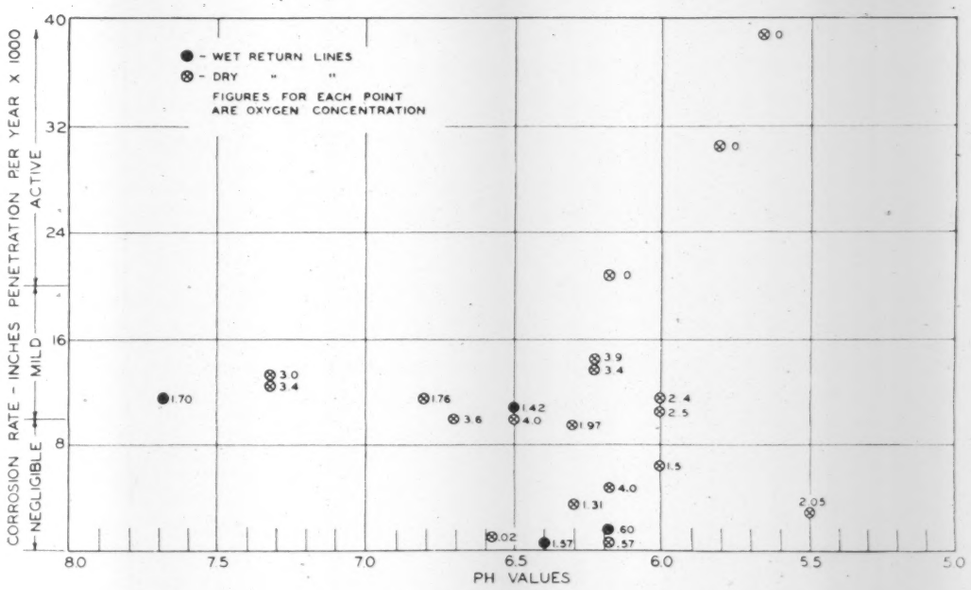


Fig. 9—Data showing the pH value of the condensates and corrosion rates found in *main* return lines of operating systems

In vacuum systems, and in pressure systems which operate intermittently, the carbon dioxide concentration of the condensates is invariably less than in the steam supply and more than can be attributed to infiltration of air. As a matter of fact, determined concentrations of CO_2 in the condensate are invariably much higher than theoretical values obtained by calculations based upon Henry's Law.* Careful analyses indicate this is not due entirely to analytical errors. It is a possibility that such concentrations of carbon dioxide do not obey the partial pressure law rigidly enough to be used as a criterion. The determination of the solubility of CO_2 in pure water at low partial pressures and at temperatures above 60 C would be a good subject for academic research.

In equipment which operates more or less continuously at positive pressures—such as water heaters, blast coils, and cookers—the steam consumption is extraordinarily high under some conditions. This tends to accumulate entrained gases in the steam space and extremely large amounts are sometimes collected. An insight into the possibilities which may eventuate in this regard was obtained by analyzing steam from the vapor space of the heater at location No. 14 for different condensation rates. These results are shown in Table 3.

Table 3—Amount of CO_2 Measured in the Vapor Space of a Water Heater at Various Condensation Rates (Location No. 14)

Condensation Rate Lb/Hr	Analyses of Vapor from Steam Space		
	Oxygen cc/L	CO_2 ppm	pH Value
0	0	166	5.4
400	0	54	5.2
1024	0	57	5.2
1280	0	59	5.2
1408	0	64	5.0
1664	0	127	4.8
1980	0	242	4.6
3000	0	610+	4.4

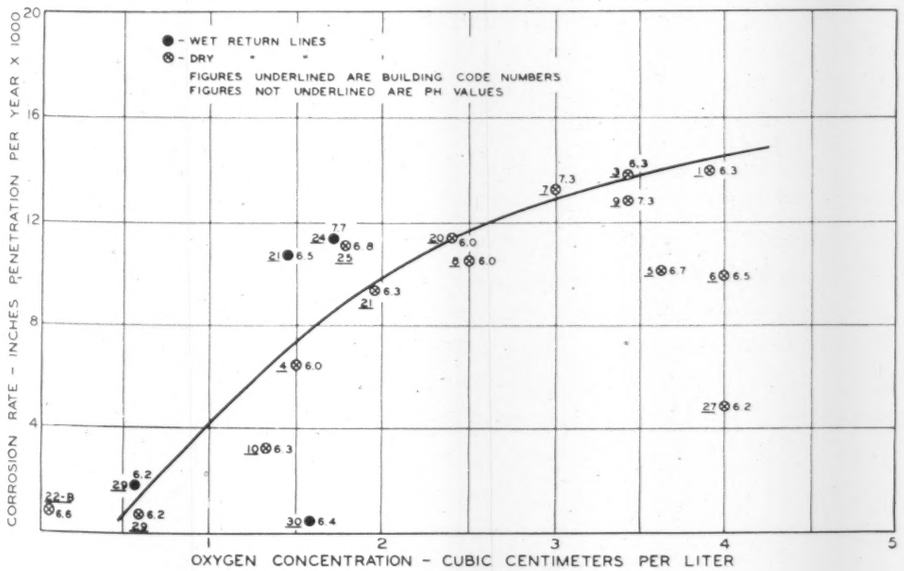
[a] CO_2 content of steam supply.

While it is theoretically possible to prevent such accumulations by venting this type of apparatus (and correspondingly to reduce corrosion as was demonstrated for this unit in Table 2) such a remedy is not altogether practical. One of the biggest difficulties is to salvage the vented steam.

[To be continued]

*Henry's Law states that the concentration of dissolved gas in solution is directly proportional to the concentration in the free space above the liquid.

Fig. 10—Corrosion as a function of oxygen concentration where pH values were 6.0 or higher



High Pressure Pipe Welding Joints ----A Review and a New Development

By H. N. Boetcher*

AS available equipment and technique, together with rigid qualification tests, for power piping welding have made it a question mainly of selection and training of men to obtain good workmanship, the joint design has assumed increasing importance. The shape of the groove, usually a "V" or a modified "U," has been standardized, though opinions still differ on a number of points such as the angle of the walls. The latter usually is a compromise between the attempt to keep down undercutting and thus assure sound fusion to parent metal and the desirability of reducing the cross-sectional area of the weld to a minimum.

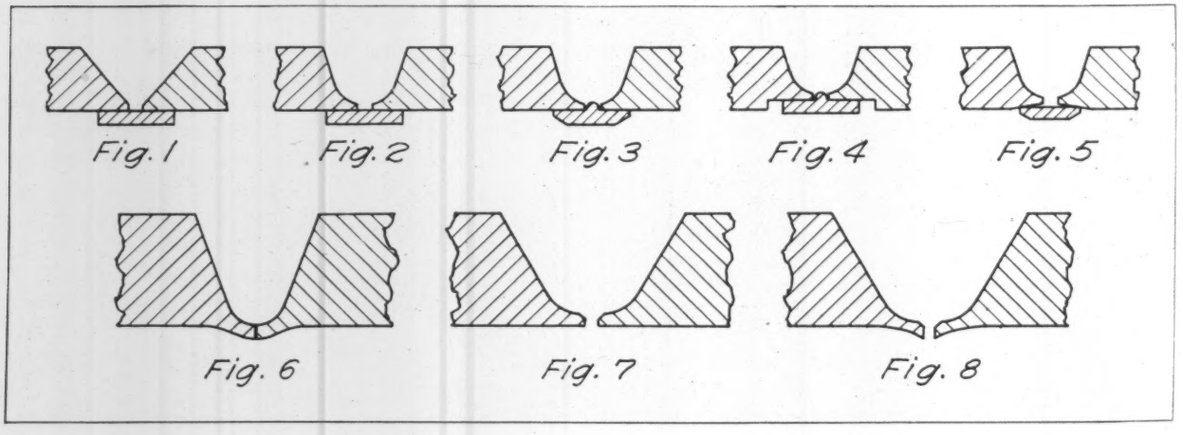
The root design is of vital importance to the soundness of a highly stressed pipe weld, and may be considered the key to a good joint—a fact which is emphasized by the multitude of designs developed and used. For good penetration, root spacing must be large enough to avoid bridging over and assure penetration to the bottom of the weld. However, with the present status of arc welding, it is not possible to obtain consistently good penetration to the root, welding into an open gap. Depending on spacing and other factors, penetration will be either insufficient, with notch effect at a vital part of the weld, or the arc will burn through. Conditions are not substantially different with gas welding. The situation is due largely to lack of control. The rapid depositing of metal by the metallic arc eliminates possibilities of correction of poor deposit along the open gap. Gas welding involves a large amount of heat needed to increase the surface temperature of the relatively heavy walls to the melting point with an accordingly large mass of fluid metal which cannot be controlled and shaped properly. Under these conditions, the gap must be closed by some means.

Separate backing rings are employed more extensively than any other method for closing the gap at the root. Excepting a few cases in which the rings are accessible for removal after welding, they remain in place, though experiments have been made to develop sectional rings of copper or other metals not fusing with steel and other types of removable rings. Rings are either continuous or split. A large number of shapes have been developed in accordance with different needs and ideas and are in use. The simplest is made of steel strip with a rectangular cross-section. To reduce resistance to flow in the pipe, the protruding ring surface is given a rounded contour in other rings. Some designs place the ring in a recess to minimize resistance to flow, with or without upsetting of pipe ends to make up for the loss of stress-carrying metal cross-section. To insure proper spacing of pipe ends, ridges and spacers are used at the outer surfaces of some rings. As such spacing members have been found to cause unfused root sections, they are in some cases removed just before welding. Since gaps between pipe and ring result in undesirable conditions in the weld, internal machining of pipe ends is usually necessary for high grade joints. This machining, in some designs, has taken the form of beveling, with the idea of obtaining improved contact between ring and pipe from contraction during welding. This is desirable as insufficient flow of heat from the ring into the pipe favors burning through the ring. The use of a combination of beveled surfaces and a spacing ridge, even if recommended by high authority[1], defeats, of course, the purpose of the beveling unless machining is carried out with impractical accuracy. Figs. 1 to 5 show some ring designs. [*Note*—In Fig. 5, the ring as well as the pipe end is beveled, although this is not clear in the sketch].

*Assistant to Superintendent of Power Production Stations, Consolidated Gas Electric Light and Power Co. of Baltimore.

[1]ASME boiler code addenda approved August 19, 1938, and August 15, 1939, Par. P-112, Fig. 5½c.

Figs. 1-5—Pipe joints with separate backing rings.
Fig. 6—A joint design involving integral backing rings.
Figs. 7-8—Sketches of two types of "Westport joint"



A critical review of conditions resulting from the use of separate backing rings indicates the reasons which prompted our attempts to get away from them.

In the first place, the backing ring constitutes a separate item which frequently has to be handled with special care to avoid its deformation and damage to machined surfaces. Second, unless machining is carried to considerable accuracy, gaps between rings and pipe ends are unavoidable. The use of continuous rings aggravates this condition. Experience has shown that "doctoring" in the field is often necessary to obtain a good enough fit. The necessity of accurate machining is increased when beveled contact surfaces are used. The advantage of beveling is lowered by tack welding of rings, as has been found necessary in many cases. Gaps between ring and pipe and sharp corners and spacers frequently are responsible for defects at the root of welds.

The use of backing rings involves either a reduction of internal pipe cross-section available for flow or a decrease of stress-carrying metal cross-section or upsetting of pipe ends to avoid such a decrease. The use of a recess results in disturbing the smooth stress flow in the pipe metal and accordingly introduces stress concentrations. The practical importance of such stress concentrations may not be known for some years. Long time experience with piping subjected to much lower stresses than used now in severe service conditions makes it appear advisable not to disregard the possibility of fatigue failures under these conditions, especially at points with unsoundness at the root. Similarly, the shape of recesses is of importance in this respect. Stress concentrations seem likely to appear also at the fusion line between ring and weld. They are caused, furthermore, by metallurgical conditions arising from the rapid changes in physical properties at the root at the point at which grain refinement from welding stops. The practical importance of this accumulation of "stress raisers" at the root of a joint made with a separate backing ring is bound to make the root, which unfortunately is the most highly stressed part of the weld, the weakest spot at the same time.

Recognition of the disadvantages offered by the separate backing rings resulted in the development (by others) of an ingenious joint involving "integral backing rings," a cross-section of which is shown in Fig. 6. In this design, the backing ring is replaced by two lips butted against each other and forming a U-groove for welding. The lips are usually rolled down as shown in Fig. 6 to insure full metal section along the throat of the weld. Conditions for welding are more favorable in this design than they are with most joints using separate backing rings and a good weld can therefore be expected at the root with greater certainty, provided that the joint is well lined up; this condition can be obtained with heavy pipe by uniform rolling of the lips.

A critical analysis of welded joint designs on the basis of stress conditions and test welds as made in preparation of welding for a high pressure extension to the Westport station at Baltimore indicated to us the desirability of further development work.

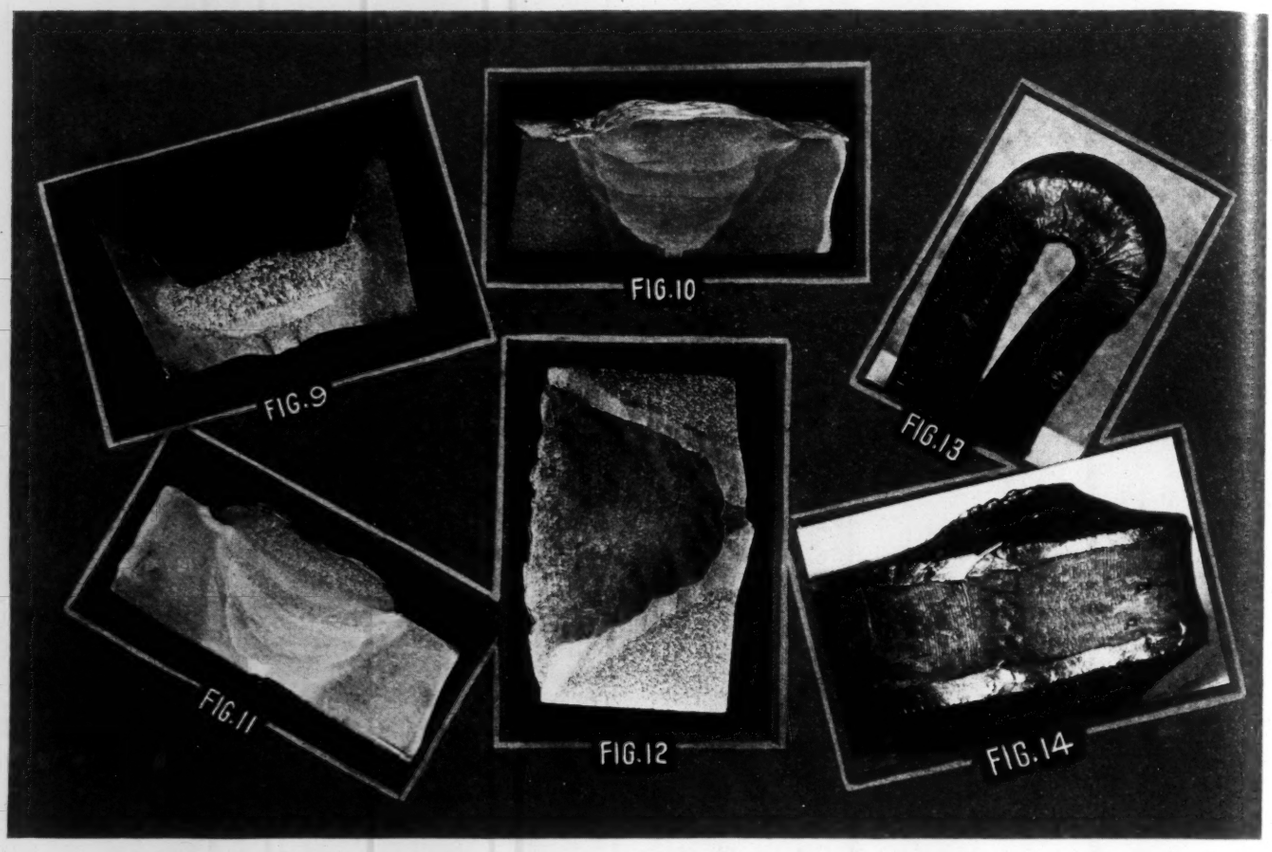
Though, for reasons already stated, the welding of heavy pipe (or plate) without closing of the gap does not give sound root conditions, thin plate has been welded successfully by different methods for a long time. Processes such as carbon arc, atomic hydrogen and oxy-acetylene welding have been used extensively with good success as they permit a greater degree of control than metallic arc welding. This fact is used in the "Westport joint" by shaping the root section into the equivalent of thin walled pipe. Oxy-acetylene welding was selected for welding the properly shaped and spaced lips at the root of the weld into a continuous integral backing ring. Apart from the small bead used for joining the lips, the weld is made by metallic electric arc welding. Figs. 7 and 8 are sketches of two types of Westport joints[2], that of Fig. 7 being used where fittings are welded to pipe and that of Fig. 8 for pipe to pipe joints. As the "backing ring" is an integral and stress carrying part of the joint, the rolling down of the lips is intended only to improve alignment and is not needed where dimensions of fittings, etc., permit satisfactory machining.

Rather extensive development work was carried out to determine the best methods to be used for the welding of the lips as well as the practicability of the joint. With the shape and spacings decided on, welding is comparatively simple. In view of the small amount of heat applied and with the use of proper technique, scaling is no more and usually less than with separate backing rings. It has been reduced still more by the use of brazing fluxes on welding rod and for pipe ends. Penetration has been found to be certain and sound. Though alignment should be carefully made as in any joint for severe service conditions, misalignment of a magnitude certain to give trouble when separate backing rings are used has been found not to affect the quality of the joint; this refers both to non-uniformity in the spacing and to radial misalignment. The training of welders in making the oxy-acetylene bead is not difficult. Good electric arc welders who had never done any gas welding before were able to make an acceptable joint after four hours of practice without instruction.

Root bend tests are the best criteria available for determining the quality of the root of a weld. A very large number of such tests were made. The specimens were cut from the pipe — usually carbon molybdenum pipe, Schedule 160—by the torch and were neither annealed after cutting nor machined in any way before testing. Bending was carried out in an American Welding Society jig. As no failures occurred in these tests, bending was afterwards continued as a free bend test in a tensile testing machine as far as possible. The tests showed high ductility of the metal at the root, as a result of the annealing effect of the gas welding and showed absence of sharp differences in physical properties in the metal. With proper selection of materials and training

[2]Patent applied for.

With improved equipment and technique, and rigid qualification tests, joint design for welded power piping installations has assumed increasing importance. A discussion of designs involving separate backing rings, and the development of a new design being used for the high and intermediate pressure piping for the Westport station extension, is given



Figs. 9-11—Cross-section of "Westport joint" in horizontal fixed pipe, Fig. 9 showing the gas and first two electric arc beads, Fig. 10 the completed weld, and Fig. 11 showing the pipe misaligned $\frac{1}{16}$ in. at inside. . . . Fig. 12—Cross-section of the joint, vertical position, misaligned. . . . Figs. 13-14—Root bend test specimens of the joint, bent to 50 per cent elongation in 1 in. across weld, side and outside views

of men, results were uniformly good. An experimental weld made with 8 in., Schedule 160 carbon molybdenum pipe in a fixed horizontal position is representative of results obtained consistently in the late stages of development work. Out of eighteen 1 in. wide specimens tested, 12 showed no cracks after elongations between 50 and 62 per cent in 1 in. across the weld; three cracked in the base metal or at the torch-cut hardened edge after elongations of 50 to 62 per cent; one had no cracks after an elongation of 44 per cent; one developed a crack in the first electric arc bead at 37 per cent; and one cracked in the fusion line in the torch-cut hardened edge at 25 per cent elongation.

With a large number of joints made by us and a piping fabricator for development and qualification purposes, it is felt that the condition at the root gives this joint a definite superiority. It seems notable that this improvement is obtained at a reduction in cost over high grade welds made with separate backing rings. The joint will be used in welding the high and intermediate pressure piping of the Westport extension now under construction.

Lithographing Plants Need Air Conditioning

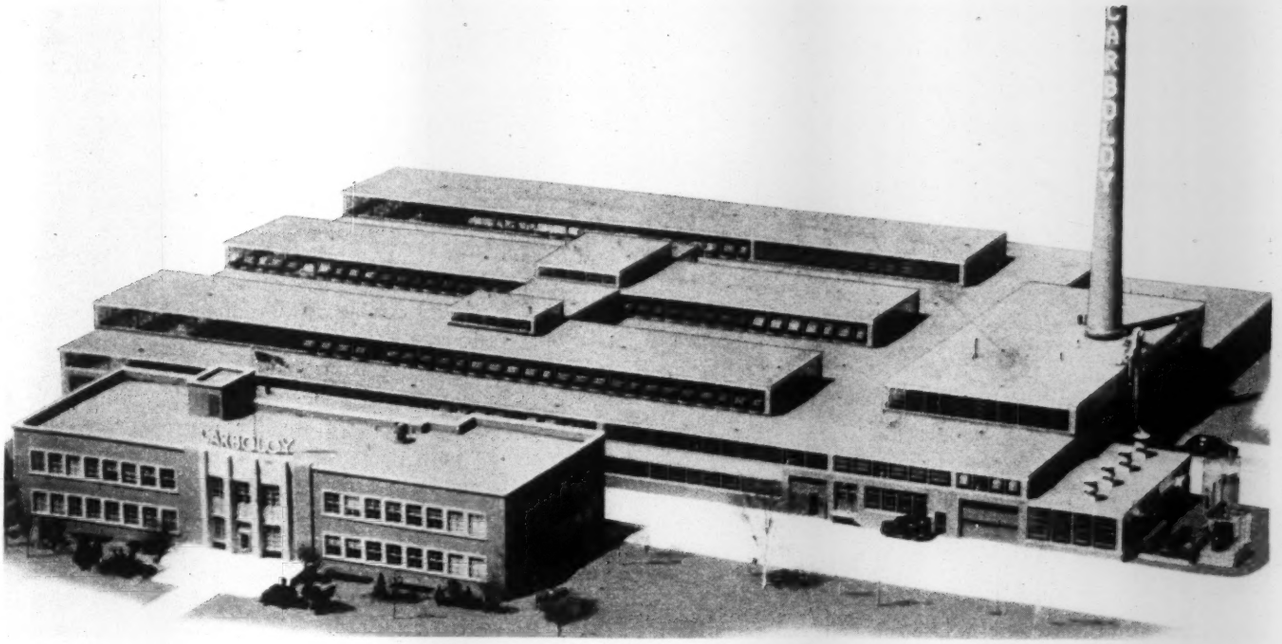
Reduction of air conditioning costs of more than 30 per cent in 10 years was cited by E. H. Dafter* before the National Association of Photo-Lithographers' convention at New York City as a reason for installing air conditioning systems "the right way" in order to save on operating costs later.

"There is no reason today to 'pinch pennies' in installing air conditioning," he explained, "because the reduced cost of equipment allows you to spread yourself. An expenditure of five or six per cent more on your 'first cost' will be returned in the first year's operation."

Pointing out that increased profits still "remain the chief reason for air conditioning," Mr. Dafter outlined the need for systems in photo-lithography plants.

"Research has shown that there are three places in the modern lithographing plant where control of atmospheric conditions is important: (1) plate making department, (2) press room, and (3) stock storage room. The plate making department calls for control of both temperature and humidity and at the same time there is a need, filled by air conditioning, for the elimination of the dirt hazard. In the press room humidity control is more important than temperature, but low humidities desired by offset printers are not practical without temperature regulation. The stock storage room should be kept at the same conditions as the press room, he said.

*Engineer, Carrier Corp.



The plant is the largest cemented carbide plant in the U. S.

Men and Metal Both Air Conditioned in New Plant of the Carboloy Company

By K. G. Patrick*

MAINTENANCE of the proper temperature and humidity to keep employees comfortable and powdered metal stocks carefully controlled for mixing has been achieved in the new $750,000 building of the Carboloy Co., Detroit, Mich., through a year 'round air conditioning system. The plant, formally opened on June 1, is the largest cemented carbide plant in the United States, and has installed facilities capable of producing approximately 10 times the amount of Carboloy currently consumed by industry. It combines the manufacturing activities formerly divided among plants in Detroit, Cleveland and Stamford, Conn.

The two story administration building, housing sales, engineering, drafting, purchasing, and accounting departments, is of reinforced concrete, completely air conditioned for summer and winter, with acoustical ceilings in halls and offices. The conditioning of the building is broken down into three distinct zones. For the "north and east" zone, an air conditioner and 40 hp condensing unit supplies 14,000 cfm of filtered and cooled or heated air, of which approximately 10 per cent is outside air. The construction of the conditioner is such that the cooling coils are segregated into two zones, each controlled by a separate thermostat, so that more even control of the temperature may be obtained. The "south and west" zone is supplied by similar equipment, but handles slightly less air.

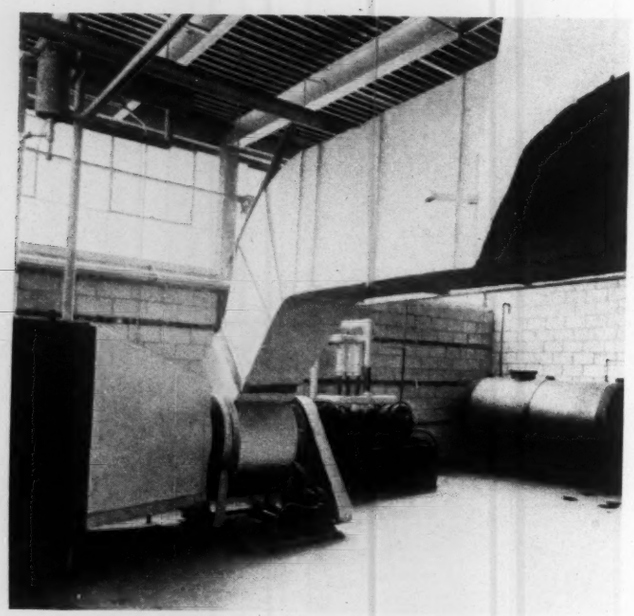
*General Electric Co.

For these zones the 40 hp condensing units supplying the cooling are equipped with automatic capacity reducing control, so that when one of the zones is satisfied, the reduced load on the coils is immediately reflected in the suction pressure at the condensing unit, and the machine shuts down to one-half capacity, effecting a saving in power consumption.

Air from the conditioners is supplied to every room through either high ceiling or high wall discharge outlets. All doors are provided with louvers so that return air passes into the hall, where it is picked up by return air ducts and sent back to the conditioner. The amount of outside air used is controlled by dampers fixed at 10 per cent during the cooling cycle, but when it is desirable to ventilate the building and completely change the air, a manual switch is provided to open the dampers wide.

The basement of the administration building, contain-

At the new plant of the Carboloy Co., Detroit, a year 'round air conditioning system maintains proper temperatures and humidities to keep employees comfortable and powdered metal stocks in the correct condition. The system and its unique control are described briefly here



Metal mixing room, showing condensing unit and conditioner

ing a recreation or assembly room and a cafeteria, is supplied by a separate conditioner and a 15 hp condensing unit, operating independently. This supplies approximately 3000 cfm of air, with provision for 50 per cent outside air if desired.

The winter operation of the system is simple, since it only supplies tempered, filtered, and humidified air to the building, the actual heating load being taken care of by concealed radiation supplied by factory steam.

The powder metal department, or "metal room" of the factory proper, is reached through air locks, installed to insure uniform operation of the air conditioning system which closely controls atmospheric conditions in this section. Constant temperature and humidity levels are maintained here for accurate weighing and blending of the powdered metals going to make up Carboloy, inasmuch as some of the materials are hygroscopic. Extreme cleanliness is also essential to prevent contamination of the powdered metals. Equipment in the room includes ball mills, crushers, sifting and mixing units. Temperature and humidity must be maintained for the entire year, on both heating and cooling cycles, at 80 F and 35 per cent relative humidity. The air conditioning system was designed to meet these requirements within 1 F and 1 per cent relative humidity. Equipment consists of an air conditioner and a 40 hp condensing unit.

The dehumidifying process is achieved automatically by the temperature of the cooling coils. The control equipment centers about a small telechron driven relay, which checks both temperature and humidity every minute. In the event that the humidity is high but no cooling effect is required, the condensing unit is started but the steam coil is also brought into operation so that too much cooling does not take place. It is also possible to humidify the air in summer if the outside humidity is below the value required.

Indicative of the economy features that have been built into the Carboloy plant is the special provision for water and metal salvage. A series of covered trenches built into the floor of the plant direct all drainage water, including rain water from the roof, and excepting that used for sanitary purposes, into a special settling tank. The water is then pumped into a cooling tower and recirculated for use in the air conditioning system, cleaning, and for general manufacturing purposes. The system also provides for the periodic salvaging of powdered carbides and other metals of value which accumulate as a result of maintenance cleaning. It is estimated that this salvaging of water will save approximately 3,000,000 gal annually.

Advantages of Air Conditioning Museums

A PREDICTION that rare books and precious art objects—now locked in air tight glass cases in most libraries and museums—will become more accessible to the public through the use of air conditioned display and reading rooms was made last month by J. F. Kooistra* in an address before the annual convention of the American Association of Museums. He said the American Library Association conducted tests at Folger Shakespearian library at Washington which indicated the main deteriorative agents of art objects are acid pollution of air, adverse temperature and humidity, light, and impurities in the paper. He quoted J. F. McCabe, building superintendent of the Art Institute of Chicago, as follows: "The lack of comprehension of the effect of humidity and temperature, as well as air filtering and uniform conditions in museums, is the greatest stumbling block in the preservation of art objects."

He cited the loss of valuable records that were stored in a museum "under such conditions they are no longer serviceable." Improper air and humidity conditions have a "recognized destructive effect upon paper, textiles, leather, wood, adhesives and some metals."

"Besides the altruistic motive of preservation of collections, economic considerations enter quite strongly into the picture," he explained. "An air conditioning system for museums should be designed for continuous operation, and will benefit the user by bringing considerable savings in maintenance of the collections themselves.

"There are two bases upon which air conditioning may be proved to be economically sound (1) considering the value of irreplaceable articles whose lives are lengthened through the maintenance of stable conditions of temperature, humidity, cleanliness and non-acidity of the air; (2) considering the actual annual dollar savings due to less cleaning required for both buildings and art objects and the greatly reduced maintenance, restoration and replacement required.

"The average size museum will spend from 15 to 20 thousand dollars annually for cleaning, maintenance and rehabilitation. Based on a rather limited experience, we believe this cost can be cut in half through the installation of complete air conditioning. Our analysis indicates that a museum can own and operate air conditioning at an annual cost equal to or less than these savings."

*Engineer, Carrier Corp.

Piping and Measurement Engineers Must Work Together for Flow Meter Accuracy

By L. K. Spink*

BENJAMIN FRANKLIN said, "We must all hang together or assuredly we shall all hang separately." No more appropriate statement could be made of the relations between the instrument or measurement engineer and the piping design engineer.

In the early days of control engineering, it was not uncommon to calculate a valve size and find it to be one-tenth of the line size or double the line size. Of course, no piping engineer would admit having installed a line without some conception of the velocity to be encountered in it, but whether these caricatures of engineering are the result of a loss of a decimal point, sheer carelessness, or a blind spot of engineering technique, they undeniably point to waste in investment costs on the one hand or a waste in pressure or in pumping or blowing costs on the other hand.

To an even greater extent, metering requirements are related to sensible piping engineering. The ratio of piping loss per 100 pipe diameters of straight pipe to the differential produced by a 0.75 d/D ratio orifice is practically constant for the non-viscous flow range. This fact is being used by many apparatus manufacturers today in the design of their piping. They find that if the pipe size is chosen to give satisfactory relations for metering, the pressure drops and pressure losses of the piping itself will take care of themselves.

Pressure Losses in Piping

Some of the data used in flow metering may be useful in promoting a better understanding of piping losses. In the comparison of the effect of sharp reductions or increases in the size of piping, a study of the pressure loss of flow nozzles, venturi tubes, and orifice plates will be of interest.

When velocity is increased, there is an accompanying decrease in pressure, due to the transfer of potential into kinetic energy. If the transfer from this high velocity back to normal velocity is slow, gentle and without turbulence, most of the pressure dropped at the point of high velocity is recovered when normal velocity is restored.

The shape of the approach section has very little effect on the pressure loss incurred due to a given change in velocity. It is true that a flow nozzle will give a lower pressure loss than an orifice plate of the same size in a line of the same size. It should be observed, however, that there is a constriction of the jet downstream from the orifice which is not present at the outlet of the flow nozzle. A flow nozzle, to give the same increase in velocity, must be smaller in diameter than the orifice plate. It will be observed that in a given line if a flow nozzle is designed to give the same pressure drop as an orifice plate, the percentage pressure loss is closely comparable. Fig. 1 shows the pressure loss of the orifice plate; Fig. 2 shows the flow nozzle, and short and long cone venturi.

It will be observed that the venturi with the short recovery cone has pressure recovery varying from more than double to approximately four times that of the flow nozzle. The long cone venturi has a pressure recovery of considerably greater than that of the venturi with a short recovery cone. The slope of the short recovery cone is one to 10, and that of the long cone venturi is one to 20.

Several conclusions of value outside the field of flow measurement may be drawn from these curves. It is evident that the shape of an enlargement of pipe section is much more important than the shape of a restriction. Enlargements, in order to result in the minimum loss of pressure, should be long and gradual. Restrictions, in order to give the minimum loss in pressure, should be nozzle-shaped, blending into tangency with the reduced section, or may, as an alternative, have the slope of the approach section of the venturi tube; one to five. Sharp reductions and, to a greater extent, sharp increases in size of pipe cause an unnecessary loss of pressure. Increases in size should, wherever possible, be made ahead of elbows or tees. Turbulence at high velocities causes unnecessary pressure loss.

Disturbed Flow

There is an unexplored field for investigation in the effect of disturbed flow on piping losses. It has been very evident to investigators in the field of orifice measurement that there is a difference between the effect of two elbows in different planes on the upstream side of

The importance of the relationship between piping design and layout and flow metering is not as generally appreciated as is necessary for accurate metering. The accuracy of meters depends very definitely upon how and where they are installed. It is Mr. Spink's purpose here to point out the things piping designers must consider, and the factors that influence meter accuracy . . . Just recently the author had a very convincing example of the importance of emphasizing this kind of information to piping engineers. He spent two days going over some 50 or 60 flow meter and flow control piping layouts which had been figured according to rule of thumb — 10 pipe diameters upstream and five downstream. Fortunately it was found that most of the bad installations could be rescued by proper use of straightening vanes

*Manager, Meter and Gage Div., The Foxboro Co.

an orifice plate and the effect of the same two elbows in the same plane as regards the pressure drop observed on the same orifice plate at the same rate of flow. Is there any reason why the relative position of elbows upstream from an orifice should radically affect the pressure drop of an orifice and not affect the pressure drop caused by a swage, valve, or even a straight run of pipe? Since pressure loss after a flow restriction or enlargement of pipe size is dependent upon smooth restoration of velocity distribution, why shouldn't a swirl producing disturbance affect this pressure recovery? The evidence at hand is not sufficient to answer these questions.

Fig. 3 shows the plotted effect of various types of flow disturbances on orifice measurement, when the disturbances are placed at various numbers of pipe diameters upstream from the orifice. The effect on the pressure drop is approximately double the plotted measurement error and is in the same direction. Straightening vanes interposed between the source of the disturbance and the measuring orifice eliminate the effects of elbows in two planes, valves, or regulators.

For instance, referring to the Buffalo disturbance tests of the joint American Gas Association-American Society of Mechanical Engineers committee, tests Nos. 70 to 75 show the following ratio coefficients of the test orifice to the reference orifice: 1.016, 1.014, 1.015, 1.015, 1.015, and 1.015. These figures represent very closely the calculated theoretical ratio of the coefficients of the two orifices in question. These tests were made with a straightening vane interposed between the source of disturbance composed of two elbows at different planes, situated at a distance of 36 ft 10 in. of 8 in. pipe upstream from the test orifice. The same test made with the straightening vane interposed between the test orifice and the two elbows in different planes at 23 ft 6 in. upstream from the orifice showed ratios for tests Nos. 90 to 94 of 1.015, 1.014, 1.014, 1.014 and 1.015.

Now take the same test set-up, except with elbows in two planes at 23 ft 6 in. upstream from the orifice and no vanes. Tests Nos. 95 to 99 inclusive showed the results of this combination. The ratios of coefficients were as follows: 1.046, 1.043, 1.042, 1.052, and 1.040.

When the elbows were placed in the same plane the measurement immediately returned to normal. Tests Nos. 131 to 133 inclusive covered the same installation with the elbows in the same plane. The ratios of coefficients were 1.016, 1.015, 1.015.

To the observers at the water columns, a very readily detectable difference was apparent when a flow disturbance productive of measurement error was present. The water column was nervous, fluctuated widely, and while the individual readings varied widely, the average of 10 readings gave quite a consistent measurement error as shown by the ratio of coefficients taken from the test data. When a measurement error was present, the flows through the pipe line were noisy. The static pressures also varied more than in the case of steady flows. The straightening vane eliminated not only the error but also the noise and the pressure fluctuation. The straightening vane has already been used by valve manufacturers to reduce the noise of reducing valves and we believe it must also have, at the same time, resulted in better pressure regulation.

Use of Straightening Vanes

Straightening vanes are simple, honeycomb units interposed in the line of flow. They consist of a number of straight parallel passages, the length of which should be approximately 10 times the maximum dimension of the opening. There should preferably be at least four of these sections across any diameter, but appreciable improvements have been obtained with straightening vanes with fewer sections. Vanes of the radial type having only two sections across the diameter have been used with reasonably satisfactory results. A properly made straightening vane has a very small percentage of shut-off area and consequently very little restriction to the free flow of an undisturbed fluid. In disturbed flows, there is some question as to whether the straightening vane does not recover more than enough of the kinetic energy of the disturbed fluid to make up for the pressure loss due to friction.

The straightening vane is effective in eliminating the error caused by any of the swirl producing disturbances used in the AGA tests. This includes combinations of elbows in two planes, partially closed valves, or regulators. These disturbances appeared to have little or no tendency to travel upstream; hence, vanes were not found to be required when the only disturbing fittings were located downstream from the measuring orifice.

When vanes are used, they should be situated at a reasonable distance from the orifice. Seven pipe diameters are sufficient for the commonly used range of orifice to pipe diameter ratios. When placed too close to the orifice, the vane itself may cause an error. The test results were within a half per cent, however, with the vanes situated within two pipe diameters of the plate. Swirl producing fittings were consistent in producing a negative error. Vanes of the types used in the tests when placed too close to the orifice caused a small error in the opposite direction. The theory has been advanced that the error due to the vanes was caused by a flattening of the velocity distribution curve.

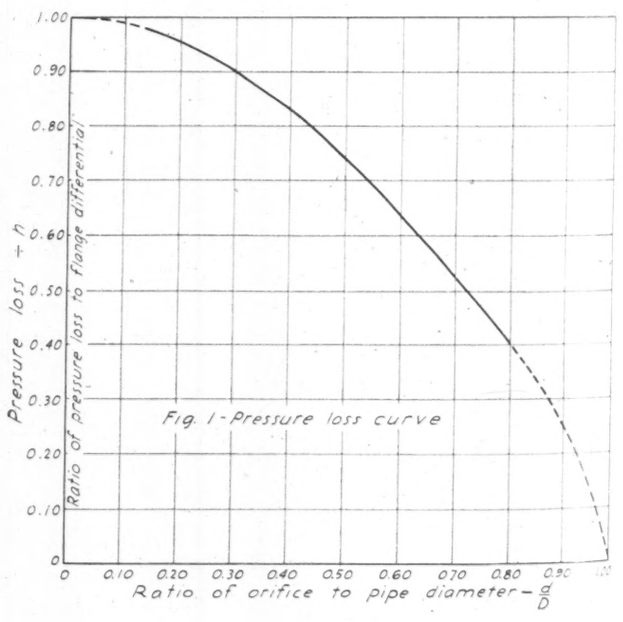


Fig. 1 - Pressure loss curve

due to the increased flow resistance at the center of the pipe.

The limitations on upstream and downstream straight run of pipe which should be placed adjacent to measuring orifices are necessarily arbitrary. The evidence of the difference between the effect of elbows in the same plane and that of elbows in two planes appeared so conclusive that the AGA gas measurement committee published the following statement in its report No. 1, in 1930, after the conclusion of the tests on disturbing fittings: "If within 100 pipe diameters upstream from the orifice plate the run of pipe includes anything other than flanges, collars, wide open gates, or bends, elbows, or tees which are all in the same plane, then a set of straightening vanes should be placed in the pipe between the upstream pressure tap and the nearest such interruption but not less than six pipe diameters away from the pressure tap." In other words, elbows in the same plane were not considered a serious disturbance.

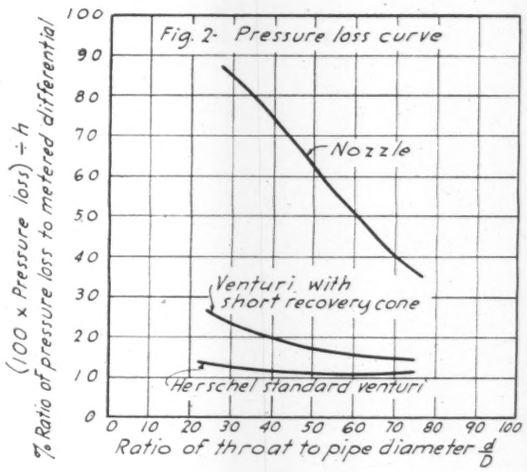
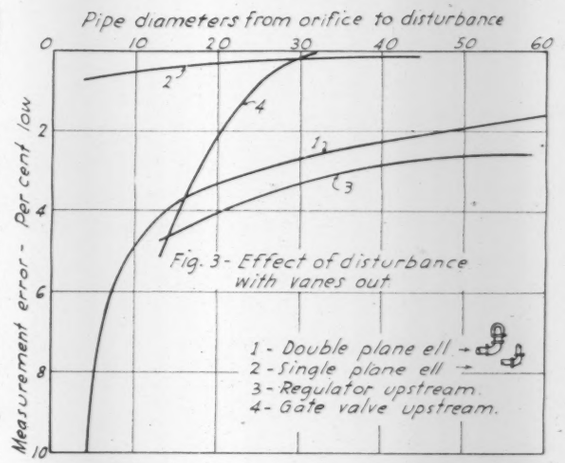


Fig. 2- Pressure loss curve
(100 × Pressure loss) ÷ h
% Ratio of pressure loss to metered differential
Nozzle
Venturi with short recovery cone
Herschel standard venturi
Ratio of throat to pipe diameter d/D


Pipe diameters from orifice to disturbance
Measurement error - Per cent low
Fig. 3- Effect of disturbance with vanes out
1 - Double plane ell
2 - Single plane ell
3 - Regulator upstream.
4 - Gate valve upstream.

Note—The curves of Fig. 3 should not be used for obtaining empirical correction factors. They cover a specific arrangement of specific fittings. The arrangement and descriptions listed on the curve are general. The amount of error from the valve or regulator definitely varied with the amount of restriction. It is unlikely that other regulators or valves would give duplicate results, and it is improbable that other combinations of elbows differently spaced would result in the same errors. [Fig. 2. Courtesy Builders Iron Foundry, data by Ed. S. Smith, Jr.]

Other publications, including the final report of the AGA committee, indicate that elbows in the same plane constitute a disturbance which justifies the use of vanes, but not as serious as the same elbows placed in different planes.

A restriction, such as a swage, situated ahead of the orifice constitutes a disturbance not covered by the foregoing discussion. It appeared evident from the tests that this type of disturbance caused a centralization of velocity giving the same effect as a reduction in the size of pipe at the orifice. The use of a straightening vane did not tend to eliminate the effects of this type of disturbance; in fact, the use of a honeycomb type of vane seemed to prolong the effect and carry it farther downstream.

Meter Accuracy Depends Upon Installation

The orifice meter consists of a primary device, the orifice plate, which constitutes a resistance to flow, and a secondary device which measures the drop in pressure created by the resistance to flow. Flow nozzles and venturi tubes are merely other types of resistance elements which may be substituted for the orifice plate to perform the same function.

The basic calibrations of all of these primary elements were made under ideal conditions. In order to use the predetermined coefficients of these primary devices, the commercial operating conditions must be the equivalent of the conditions under which the laboratory calibrations were made.

Some conditions, such as the straight run of pipe preceding and following the orifice or primary device, may be varied through wide limits without appreciably affecting the results. Other conditions must be duplicated within close limits.

Ratio of Orifice to Pipe Size—The accuracy with which conditions must be duplicated and hence, the delicacy of measurement by the flow meter are strongly dependent upon the ratio of orifice to pipe size. For good practice, a reasonable limit of 0.75 or preferably 0.70 should be placed on this ratio. If the velocity of flow is high, a high ratio of orifice to pipe size is necessary to bring the differential pressures down in a usable range. The usable differential pressure is limited, on compressible fluids, by the static pressure. For best practice, the differential pressure created by the orifice plate, in inches of water, should not exceed the absolute static pressure in pounds per square inch. This limitation together with the limit on orifice size governs the maximum allowable velocity in the meter run. Somewhat higher velocities can be handled by substituting a flow nozzle for the orifice plate, but this alternative has obvious disadvantages. Consequently, it is desirable to choose a pipe size large enough to permit accurate measurement with the orifice plate.

Approach Piping—On high ratios of orifice to pipe size, the exact internal diameter of the approach piping is extremely important. Normal variations in commercial pipes make it impossible to maintain strict tolerances; hence, the selection of smooth, round, accurately dimensioned pipes is necessary for the fabrication of meter runs. As the ratio of orifice to pipe size decreases, the exact characteristics of the approach pipe become less important; the accurately bored orifice plate becomes the controlling factor. On low ratios, appreciable accumulations of sediment or corrosion may occur on the inside of the pipe without appreciably affecting the readings of the meter.

The allowable tolerances placed on diameter of pipe by the AGA committee reflect the increasing delicacy of measurement as the orifice to pipe ratio increases. For a diameter ratio of 10 per cent to 25 per cent, with close-up taps, this report recommends a tolerance of 5

per cent; from 25 per cent to 35 per cent, a tolerance of 3 per cent; 35 per cent to 45 per cent, a tolerance of 2 per cent; from 45 per cent to 55 per cent, a tolerance of 1½ per cent; from 55 per cent to 65 per cent, a tolerance of 1 per cent; 65 per cent to 75 per cent, a tolerance of ½ per cent.

It is common knowledge that rough or tubercular pipes can give a flow resistance equivalent to that of a much smaller pipe. This is also true of its effect on flow measurement. In that case, it results in a centralization of velocity, which reduces the relative restriction of the orifice plate.

Steel tubing is being widely used for orifice meter runs. Several manufacturers are making it to extremely close tolerances. Spiral, riveted, and similarly fabricated pipes should be avoided in the construction of meter runs where maximum accuracy is desired. Slight projections on the inside of the pipe in the immediate neighborhood of the pressure taps may create an error far in excess of the actual reduction in pipe area, due to the suction or impact effects they may produce on the pressure taps. The pressure tap hole should be accurately situated, and, around the inner surface of the pipe, should be free from burrs and slightly rounded. The diameter of the hole where it enters the pipe should not be greater than one-eighth of the inside diameter of the pipe.

Orifice Meter Flange—The installation of the orifice meter flange, in which the plate is to be located, is important. The pipe should be smooth, clean, concentric, and of uniform accurate diameter up to within ¼ in. of the face of the flange. Pipe cutter burrs or welding icicles should be carefully removed from the end of the pipe. Particular attention should be paid to the pipe at the entrance to the flange. The use of pipe containing bumps or irregularities near the location of the pressure taps should be avoided. Flange gaskets should be trimmed and positioned so they cannot extend into the opening of the pipe. If the measured gas or steam contains condensate, provision should be made to prevent it from piling up against the orifice plate. This is frequently done by drilling a small drain hole flush with the bottom of the line. Installations in vertical lines with the flow in a downward direction are particularly recommended where large deposits of condensate are likely to occur.

In the measurement of liquids the damming of gas or vapor in the top of the line ahead of the orifice should be avoided in a similar manner. The drain hole in that case should be placed flush with the top of the line, and if an orifice location in the vertical pipe is selected, it should be placed where the flow is in an upward direction.

Segmental or eccentric orifice plates are frequently useful in handling problems of this type. They should, in general, be avoided wherever possible, because the large volume of data on the performance of concentric orifice plates under various conditions is not available for either of these types of primary devices.

Instrument Location—On liquid and steam flow meters, more satisfactory operation will be assured if the instrument can be mounted below the orifice with a good steep slope on the piping. This provides an automatic means of disposing of air, gas, or vapors in the meter or piping through the pressure taps into the line of the flowing fluid. This does not mean that satisfactory and trouble-free installations cannot be made with the instrument situated 200 or 300 ft from the orifice. There are many installations in existence with piping leads more than 200 ft in length that have given no operation troubles from this source, but such installations require large diameter connecting pipe, properly sloped, and carefully filled to eliminate air or vapors.

Equally satisfactory installations can be made on dry air or gas with the instrument above or below the orifice. If the air or gas contains condensable vapors or condensate in suspension, better results will be obtained if the piping is arranged to drain any condensate formed or deposited in it back into the line of flow.

The Orifice

To return to the more general discussion of flow meters with round concentric orifices, it should be pointed out that the upstream edge of the measuring orifice should be square, clean, and sharp. The upstream edge should not appreciably reflect a beam of light when viewed without magnification. In the smaller sizes, the sharpness of the edge becomes increasingly important.

The upstream face of the orifice should be flat within 0.01 in. per in. of pipe radius. The orifice plate should be thick enough to maintain this flatness when placed in the pipe line under the maximum differential to which it will be subjected, but the thickness at the orifice edge should not exceed one-thirtieth of the pipe diameter, one-eighth of the orifice diameter or one-fourth of the difference between the radius of the orifice and the radius of the pipe. If a thicker orifice is required for rigidity, the thickness at the orifice bore may be reduced by beveling or counterboring. If beveled, the angle of the bevel should not be more than 45 deg from the face of the plate. Beveling should always be done on the downstream side.

In conclusion, let us summarize some of the more important points which have been brought out in the foregoing discussion:

1) Piping at the location of the metering orifice should be made large enough to give a reasonable ratio of orifice to pipe diameter. This ratio should not exceed 0.75 and should preferably be under 0.7.

2) Sufficient straight run of pipe should be allowed upstream and downstream from the proposed location of the measuring orifice, with due regard being given to whether or not straightening vanes will be used.

3) Pipe for the meter run should be selected on the basis of the expected ratio of orifice to pipe diameter. It should be remembered that the approach piping is definitely a part of the orifice meter.

4) Extraordinary care should be observed in installing the orifice meter flange. The pipe should be smooth and of uniform accurate diameter up to within ¼ in. of the face of the flange.

5) Provision should be made to prevent damming up of foreign material of different density from that of the flowing fluid ahead of the orifice plate.

6) The orifice location should be chosen with due consideration to the respective location of the instrument, to provide for a favorable arrangement of the connecting piping.

7) The orifice plate should be sharp, flat, rigid, and not greater than the specified thickness. Any counterboring or beveling should be done on the downstream side.

Combustion Space and Setting Height for Stoker Fired Boilers Is Studied

A FIELD survey on the relation of combustion space and setting height to smoke when firing bituminous coal with underfeed stokers has recently been completed, the research work being conducted by engineers of Battelle Memorial Institute for the Stoker Manufacturers Association, the National Coal Association, the Institute of Boiler and Radiator Manufacturers, and the Steel Heating Boiler Institute. The Smoke Prevention Association coöperated by the appointment of two of its members to the general steering committee in charge of the project, and as reported in the July HPAC, the report was first presented as a paper in abstract form by R. C. Cross, R. A. Sherman, and H. N. Ostborg, engineers of Battelle, at the SPA's annual meeting in Milwaukee.

The complete report has now been approved by the steering committee and is ready for release.* It presents the data obtained in a factual survey of 102 installations and an experimental study of 22 installations in Columbus, Cincinnati, St. Louis, and Chicago. The survey included both high and low volatile coals and cast iron and steel boilers fired with stokers burning 20 to 900 lb per hr of coal. The smoke emissions were recorded continuously by a photo-electric instrument; in addition, the temperature and composition of the flue gases, the rate of coal feed, the drafts and pressures, and attention to the stoker were recorded. The coal used was sampled and analyzed. Complete data on the construction and dimensions of the furnaces and boilers were obtained.

Significance of the Data

The data obtained in the general survey of 102 installations and the experimental survey of 22 installations lead to the following preliminary conclusions, which—with the final conclusions—are quoted from the report:

1) The amount of smoke emitted from the stacks could not be considered serious from the standpoint of smoke abatement ordinances as only three of the 22 installations gave periods of violation.

2) The emission of smoke was not related to the rate of heat release and only slightly to the setting height of the stoker.

3) The amount of smoke emitted in the "off" periods was greater than that emitted during "on" periods. There were but three exceptions to this general rule in the 22 test installations.

The lack of any definite relation between smoke emission and rate of heat liberation and the indication of only a slight trend of relation of smoke emission to setting height may appear surprising in view of the generally accepted belief, which was shared by the authors, that such relations exist. The conclusions have been drawn for the moment only for the installations included in the survey and critics might object to the extension of these conclusions to the some 400,000 stoker installations of the country on the grounds (1) that the number of installations sampled was too few, and (2) that such important factors as type of boiler, type of stoker, type of coal, and operating and load conditions varied too greatly to establish any relations or trends.

The number of installations was admittedly small, but recent experience with other types of surveys has established the fact that not the number of units taken for a sample, but the accuracy of the sample determines the accuracy of the conclusions applied to the whole. It is believed that the data relative to the installations surveyed show them to be so representative of stoker installations that they can be considered an accurate sample. The second possible objection relative to lack of uniformity of conditions will be considered later.

Experimental investigations of the relation of combustion space or setting heights to smoke emission have been lacking, but several investigations of the relation of combustion space to the efficiency of combustion of coal have been made. Typical of these are some of the early investigations of the Bureau of Mines, which clearly established the importance of ample combustion volume in efficient combustion. It is probable that we have all jumped to the conclusion that similar relations hold for smoke emission. In view of the data obtained in the present investigation a reconsideration of the theories of the relation of smoke emission to combustion space is in order.

The fact is well established that soot or smoke does not result from the preferential combustion of the hydrogen in a hydrocarbon fuel such as gas, oil, or coal, but from decomposition or cracking of the hydrocarbon. If a certain minimum ratio of oxygen is mixed with the hydrocarbon at the time of ignition, the liberation of carbon as soot is prevented.

This can be well illustrated with an ordinary bunsen burner and manufactured gas, or better, with an unsaturated hydrocarbon that is more readily cracked, such as acetylene. If acetylene is burned in a bunsen or inspirating type burner and at least one volume of oxygen for each volume of acetylene is admitted through the air ports, which is only 40 per cent of the amount required for complete combustion, the flame will be non-luminous. No soot is liberated and it would be impossible to have smoke in the combustion of such a mixture no matter how limited the combustion space nor how low the setting height. Combustion might be incomplete as shown by the presence of hydrogen and carbon monoxide in the gases, but there would be no smoke.

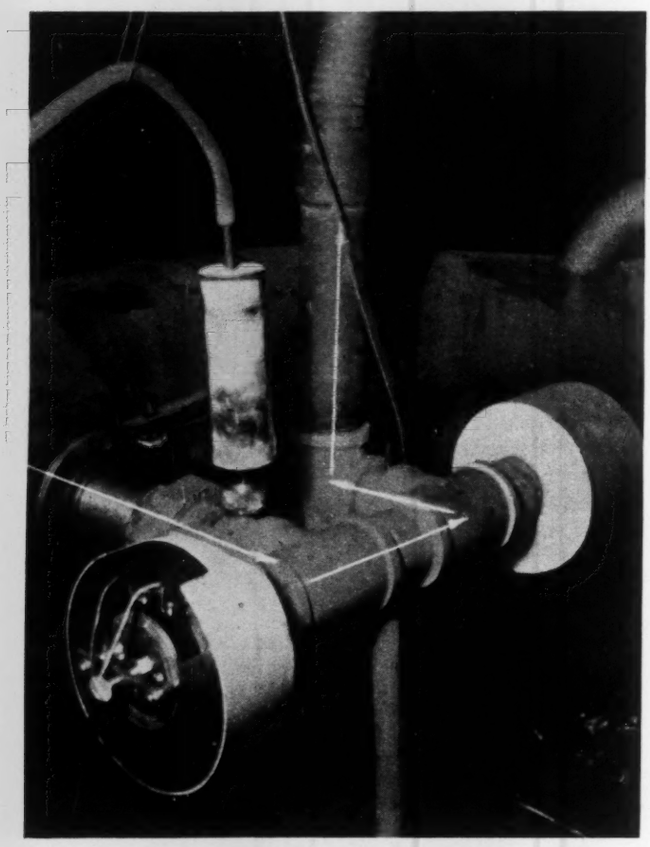
Ideally, the underfeed stoker should produce this type of combustion because the air is admitted through tuyeres at the level at which the volatile is released from the coal. The mixture is not perfect, however, and the flame is always luminous. That is, the flame contains free carbon that resulted from the decomposition of the hydrocarbons. The flame is much like that obtained if

*A Field Survey of the Relation of Combustion Space and Setting Height to Smoke When Firing Bituminous Coal with Underfeed Stokers, by R. C. Cross, R. A. Sherman, and H. N. Ostborg, 33 pp., 8½ x 11 in., mimeographed. Available from the Stoker Manufacturers Association, 307 N. Michigan Ave., Chicago, Ill., for $1.00.

we restrict the amount of air supplied to our acetylene burner. The flame becomes luminous but the carbon is burned and no visible smoke is emitted.

If, however, the flame impinges on a cold surface such as the tubes or crown sheet of a boiler, combustion will be stopped, carbon will be liberated, and we shall have smoke. The setting height and combustion space must be great enough to prevent flames from striking cold surfaces.

The amount of smoke produced in this manner is much less, however, than that produced if we cut off the primary air completely from our acetylene burner.



The smoke sampler used in the tests. The arrows indicate direction of smoke travel

The only air that reaches the gas then is that which enters by diffusion from the surrounding atmosphere; much of the gas is cracked and large flakes of soot separate. This is what happens when the distribution of coal is imperfect in the stoker, when the fire is sliced or hand fired, and air does not reach the volatile, or when the stoker shuts off. Volatile continues to be released, little air is available, and the velocity at which it enters is so low that mixing is poor.

Smoke of this type is very difficult to burn. As Kreisinger, Augustine, and Ovitz, in Bulletin 135 of the Bureau of Mines, say: "Experience with burning soft coal shows that if soot is once formed a large percentage remains floating in the gases after all other gaseous combustible has been almost completely burned. . . . Even a very large combustion space is not an unfailing solution for burning soot without a large excess of air."

The only solution for imperfect distribution of coal is a redesign of the stoker or the installation of high velocity steam-air jets directed at the offending area. By mixing of air with the gases, the formation of smoke may be avoided. Too much emphasis cannot be placed on the fact that the way to prevent smoke emission from chimneys and stacks is to prevent its formation at the fuel bed rather than to let it form and then to attempt to burn it in a large combustion space and by a high excess of air.

Let us now return to the second criticism that might be made of this work, namely, that the conditions were so variable that no general conclusions can be drawn. This is admitted and the need of a further laboratory study in which conditions can be controlled and variables changed one at a time is obvious if accurate determinations of the relation of the setting height to smoke emission are to be made. But the very fact that these variables, such as type of coal, type of load, and excess air, are important, proves that no one is in a position to state that a given rate of heat liberation shall not be exceeded nor that the setting height shall not fall short of a certain value and be assured that smoke emission will not be serious.

Conclusions

The results of this preliminary field survey of the relation of combustion space and setting heights for bituminous coal fired by underfeed stokers lead to the following general conclusions:

1) Only three of the 22 installations tested emitted smoke dense enough to be considered a violation.

2) No relation was evident between rate of heat liberation and smoke emission.

3) A trend toward less smoke with increased setting height was found for stokers feeding more than 400 lb per hr of coal.

4) The principle of the underfeed stoker meets the requirements for smokeless combustion in that the formation of smoke is prevented by the admission of air at the level where the volatile is released from the coal. This principle does not hold when the distribution of coal is non-uniform and when the stoker shuts off.

5) The total smoke emission in the "off" periods of the stoker was greater than in the "on" periods in 19 of the 22 installations tested. No amount of combustion space or setting height would prevent this smoke. The creation of a method to prevent this smoke emission is a distinct challenge to the manufacturers of stokers.

6) Other factors such as type of coal, uniformity of distribution of coal on the stoker, excess air, type of load, and degree of intelligence and ability of the operator are of equal or greater importance in elimination of smoke than the rate of heat liberation or setting height.

7) A laboratory investigation where the variable factors can be controlled is desirable to establish more definite correlations between smoke emission and setting height.

Unloading Liquid Caustic

Manual sheet TC-3 of the Manufacturing Chemists' Association of the United States, 608 Woodward Bldg., Washington, D. C., covers the unloading of steel tank cars filled with liquid caustic soda or caustic potash, and is a recommended practice of the Association, from whom it may be obtained for 10 cents.

Information is given on placement of car for unloading, safety rules, cold weather handling—steaming, sampling, unloading of caustic cars, preparation of empty car for return, materials and equipment (piping, joints, valves, gaskets and pumps), delivery lines, etc.

Properties of Mixtures of Air and Saturated Water Vapor for Barometric Pressures from 22 In. to 32 In. of Mercury

By William Goodman*

Barometric Pressure 26 In. of Mercury

All tabulated values (except temperatures and pressures) are quantities per pound of dry air in the mixture.

Temp Deg Fahr	Absolute Humidity		Volume		Humid Specific Heat	1061w	Enthalpy (Heat Content)		Temp Deg Fahr	Absolute Humidity		Volume		Humid Specific Heat	1061w	Enthalpy (Heat Content)	
	Pounds	Grains	Dry Air Cu Ft	Volume Increase of Humid Air Per Cent			Water Vapor Btu	Air-Vapor Mixture Btu		Pounds	Grains	Dry Air Cu Ft	Volume Increase of Humid Air Per Cent			Water Vapor Btu	Air-Vapor Mixture Btu
t	w	w	v_D		j''		h_v	h	t	w	w	v_D		j''		h_v	h
1	2	3	4	5	6	7	8	9	1	2	3	4	5	6	7	8	9
−40	0.0001339	0.9374	12.17	0.02153	0.2401	0.1421	0.1397	−9.460	40	0.005988	41.92	14.50	0.9623	0.2427	6.354	6.463	16.06
−39	.0001418	0.9924	12.20	.02279	.2401	.1504	.1479	−9.212	41	.006227	43.59	14.52	1.001	.2428	6.607	6.723	16.56
−38	.0001502	1.051	12.23	.02415	.2401	.1594	.1568	−8.963	42	.006475	45.33	14.55	1.040	.2429	6.870	6.994	17.07
−37	.0001590	1.113	12.26	.02557	.2401	.1687	.1661	−8.714	43	.006731	47.11	14.58	1.082	.2430	7.141	7.274	17.59
−36	.0001683	1.178	12.29	.02706	.2401	.1786	.1759	−8.464	44	.006996	48.97	14.61	1.125	.2432	7.422	7.561	18.12
−35	0.0001780	1.246	12.32	0.02861	0.2401	0.1888	0.1860	−8.214	45	0.007274	50.92	14.64	1.169	0.2433	7.718	7.867	18.67
−34	.0001884	1.319	12.35	.03028	.2401	.1999	.1970	−7.963	46	.007559	52.91	14.67	1.215	.2434	8.070	8.178	19.22
−33	.0001991	1.394	12.38	.03201	.2401	.2113	.2083	−7.712	47	.007853	54.97	14.70	1.262	.2435	8.332	8.500	19.78
−32	.0002106	1.474	12.41	.03386	.2401	.2234	.2204	−7.460	48	.008159	57.11	14.73	1.311	.2437	8.656	8.834	20.35
−31	.0002224	1.557	12.44	.03575	.2401	.2359	.2329	−7.207	49	.008474	59.32	14.76	1.362	.2438	8.991	9.179	20.94
−30	0.0002351	1.646	12.46	0.03780	0.2401	0.2494	0.2463	−6.954	50	0.008804	61.63	14.79	1.414	0.2440	9.341	9.540	21.54
−29	.0002483	1.738	12.49	.03992	.2401	.2635	.2603	−6.700	51	.009144	64.01	14.81	1.469	.2441	9.702	9.913	22.15
−28	.0002620	1.834	12.52	.04212	.2401	.2780	.2747	−6.445	52	.009489	66.42	14.84	1.525	.2443	10.07	10.29	22.77
−27	.0002767	1.937	12.55	.04448	.2401	.2935	.2902	−6.190	53	.009857	69.00	14.87	1.583	.2444	10.46	10.69	23.41
−26	.0002919	2.043	12.58	.04692	.2401	.3097	.3063	−5.934	54	.01023	71.58	14.90	1.643	.2446	10.85	11.10	24.06
−25	0.0003078	2.155	12.61	0.04949	0.2401	0.3266	0.3232	−5.677	55	0.01061	74.29	14.93	1.705	0.2448	11.26	11.52	24.72
−24	.0003247	2.273	12.64	.05220	.2402	.3445	.3410	−5.419	56	.01101	77.10	14.96	1.769	.2450	11.69	11.97	25.41
−23	.0003425	2.398	12.67	.05506	.2402	.3634	.3599	−5.160	57	.01143	79.98	14.99	1.836	.2451	12.12	12.42	26.10
−22	.0003609	2.526	12.70	.05802	.2402	.3829	.3793	−4.901	58	.01186	82.99	15.02	1.904	.2453	12.58	12.89	26.81
−21	.0003803	2.662	12.73	.06113	.2402	.4034	.3999	−4.640	59	.01229	86.04	15.05	1.975	.2455	13.04	13.37	27.54
−20	0.0004007	2.805	12.75	0.06442	0.2402	0.4251	0.4215	−4.379	60	0.01274	89.21	15.08	2.048	0.2457	13.52	13.87	28.27
−19	.0004221	2.955	12.78	.06787	.2402	.4479	.4443	−4.116	61	.01322	92.52	15.10	2.124	.2460	14.02	14.39	29.03
−18	.0004443	3.110	12.81	.07143	.2402	.4714	.4678	−3.852	62	.01371	95.98	15.13	2.202	.2462	14.55	14.93	29.81
−17	.0004676	3.273	12.84	.07517	.2402	.4961	.4926	−3.588	63	.01421	99.50	15.16	2.282	.2464	15.08	15.48	30.60
−16	.0004924	3.447	12.87	.07916	.2402	.5224	.5188	−3.321	64	.01473	103.1	15.19	2.366	.2466	15.62	16.05	31.41
−15	0.0005177	3.624	12.90	0.08323	0.2402	0.5493	0.5458	−3.054	65	0.01527	106.9	15.22	2.452	0.2469	16.20	16.65	32.25
−14	.0005446	3.812	12.93	.08755	.2403	.5778	.5743	−2.786	66	.01582	110.7	15.25	2.540	.2471	16.78	17.25	33.09
−13	.0005728	4.010	12.96	.09209	.2403	.6078	.6045	−2.516	67	.01639	114.8	15.28	2.633	.2474	17.39	17.89	33.97
−12	.0006021	4.215	12.99	.09680	.2403	.6389	.6356	−2.244	68	.01698	118.9	15.31	2.727	.2476	18.02	18.54	34.86
−11	.0006325	4.427	13.02	.1017	.2403	.6710	.6679	−1.972	69	.01759	123.1	15.34	2.825	.2479	18.66	19.21	35.77
−10	0.0006651	4.656	13.04	0.1069	0.2403	0.7057	0.7027	−1.697	70	0.01822	127.6	15.37	2.926	0.2482	19.33	19.90	36.70
−9	.0006983	4.888	13.07	.1123	.2403	.7409	.7381	−1.422	71	.01887	132.1	15.39	3.030	.2485	20.02	20.62	37.66
−8	.0007339	5.137	13.10	.1180	.2403	.7787	.7760	−1.144	72	.01954	136.8	15.42	3.139	.2488	20.74	21.37	38.65
−7	.0007700	5.390	13.13	.1238	.2404	.8170	.8146	−0.8654	73	.02024	141.7	15.45	3.250	.2491	21.47	22.13	39.65
−6	.0008081	5.657	13.16	.1299	.2404	.8574	.8552	−0.5848	74	.02095	146.7	15.48	3.365	.2494	22.23	22.92	40.68
−5	0.0008487	5.941	13.19	0.1364	0.2404	0.9004	0.8986	−0.3014	75	0.02169	151.8	15.51	3.482	0.2498	23.01	23.74	41.74
−4	.0008902	6.231	13.22	.1431	.2404	0.9445	0.9429	−0.0171	76	.02245	157.1	15.54	3.604	.2501	23.82	24.58	42.82
−3	.0009337	6.536	13.25	.1501	.2404	0.9906	0.9894	+0.2694	77	.02324	162.7	15.57	3.732	.2505	24.66	25.46	43.94
−2	.0009796	6.857	13.28	.1575	.2404	1.039	1.039	0.5585	78	.02405	168.4	15.60	3.862	.2508	25.52	26.36	45.08
−1	.001028	7.193	13.31	.1652	.2405	1.090	1.090	0.8498	79	.02489	174.2	15.63	3.996	.2512	26.40	27.28	46.24
0	0.001076	7.535	13.33	0.1731	0.2405	1.142	1.142	1.142	80	0.02575	180.2	15.66	4.133	0.2516	27.32	28.23	47.43
1	.001128	7.894	13.36	.1813	.2405	1.197	1.197	1.437	81	.02664	186.5	15.68	4.278	.2520	28.27	29.23	48.67
2	.001182	8.271	13.39	.1900	.2405	1.254	1.255	1.735	82	.02756	192.9	15.71	4.424	.2524	29.24	30.25	49.93
3	.001238	8.664	13.42	.1990	.2406	1.313	1.315	2.035	83	.02851	199.6	15.74	4.577	.2528	30.25	31.30	51.22
4	.001297	9.078	13.45	.2085	.2406	1.376	1.378	2.338	84	.02948	206.4	15.77	4.732	.2533	31.28	32.38	52.54
5	0.001358	9.503	13.48	0.2182	0.2406	1.440	1.443	2.643	85	0.03049	213.4	15.80	4.894	0.2537	32.35	33.50	53.90
6	.001421	9.944	13.51	.2284	.2406	1.507	1.511	2.951	86	.03154	220.8	15.83	5.061	.2542	33.46	34.66	55.30
7	.001487	10.41	13.54	.2390	.2407	1.577	1.582	3.262	87	.03261	228.3	15.86	5.233	.2547	34.60	35.86	56.74
8	.001556	10.89	13.57	.2501	.2407	1.651	1.656	3.576	88	.03371	236.0	15.89	5.410	.2552	35.77	37.08	58.20
9	.001628	11.40	13.60	.2618	.2407	1.727	1.734	3.894	89	.03486	244.0	15.92	5.595	.2557	36.99	38.37	59.73
10	0.001702	11.92	13.62	0.2737	0.2408	1.806	1.814	4.214	90	0.03604	252.3	15.95	5.783	0.2562	38.24	39.68	61.28
11	.001781	12.47	13.65	.2864	.2408	1.890	1.899	4.539	91	.03726	260.8	15.97	5.979	.2568	39.53	41.04	62.88
12	.001862	13.04	13.68	.2994	.2408	1.976	1.986	4.866	92	.03850	269.5	16.00	6.178	.2573	40.85	42.42	64.50
13	.001945	13.61	13.71	.3126	.2409	2.063	2.075	5.195	93	.03980	278.6	16.03	6.386	.2579	42.23	43.87	66.19
14	.002034	14.24	13.74	.3271	.2409	2.158	2.171	5.531	94	.04114	288.0	16.06	6.600	.2585	43.65	45.36	67.92
15	0.002126	14.88	13.77	0.3417	0.2410	2.255	2.270	5.870	95	0.04251	297.6	16.09	6.820	0.2591	45.10	46.89	69.69
16	.002220	15.54	13.80	.3569	.2410	2.355	2.371	6.211	96	.04392	307.5	16.12	7.047	.2598	46.60	48.47	71.51
17	.002318	16.23	13.83	.3726	.2410	2.459	2.477	6.557	97	.04538	317.6	16.15	7.280	.2604	48.14	50.09	73.37
18	.002420	16.94	13.86	.3891	.2411	2.568	2.588	6.908	98	.04690	328.3	16.18	7.524	.2611	49.76	51.79	75.31
19	.002527	17.69	13.89	.4063	.2411	2.682	2.703	7.263	99	.04846	339.2	16.21	7.774	.2618	51.41	53.54	77.30
20	0.002637	18.46	13.91	0.4239	0.2412	2.798	2.822	7.622	100	0.05005	350.3	16.24	8.029	0.2625	53.10	55.31	79.31
21	.002754	19.28	13.94	.4427	.2412	2.922	2.948	7.986	101	.05171	362.0	16.26	8.294	.2633	54.86	57.17	81.41
22	.002872	20.10	13.97	.4617	.2413	3.047	3.075	8.355	102	.05342	373.9	16.29	8.568	.2640	56.68	59.09	83.57
23	.002996	20.97	14.00	.4817	.2414	3.179	3.210	8.730	103	.05519	386.3	16.32	8.848	.2648	58.55	61.06	85.78
24	.003125	21.88	14.03	.5024	.2414	3.316	3.350	9.110	104	.05700	399.0	16.35	9.142	.2657	60.48	63.10	88.06
25	0.003257	22.78	14.06	0.5236	0.2415	3.456	3.492	9.492	105	0.05886	412.1	16.38	9.440	0.2665	62.46	65.18	90.38
26	.003398	23.79	14.09	.5463	.2415	3.605	3.645	9.885	106	.06080	425.6	16.41	9.751	.2674	64.51	67.35	92.79
27	.003541	24.78	14.12	.5692	.2416	3.757	3.800	10.28	107	.06281	439.7	16.44	10.07	.2683	66.64	69.60	95.28
28	.003689	25.83	14.15	.5931	.2417	3.915	3.961	10.68	108	.06486	454.0	16.47	10.40	.2692	68.81	71.90	97.82
29	.003845	26.91	14.18	.6181	.2417	4.079	4.130	11.09	109	.06698	468.9	16.50	10.74	.2701	71.07	74.29	100.45
30	0.004006	28.04	14.20	0.6441	0.2418	4.250	4.305	11.51	110	0.06918	484.3	16.53	11.09	0.2711	73.40	76.75	103.15
31	.004175	29.22	14.23	.6712	.2419	4.430	4.488	11.93	111	.07142	500.0	16.55	11.45	.2721	75.78	79.27	105.91
32	.004344	30.41	14.26	.6983	.2420	4.609	4.674	12.35	112	.07376	516.4	16.58	11.83	.2732	78.26	81.90	108.78
33	.004527	31.69	14.29	.7276	.2420	4.804	4.872	12.79	113	.07618	533.3	16.61	12.21	.2743	80.83	84.61	111.73
34	.004714	33.00	14.32	.7578	.2421	5.002	5.076	13.24	114	.07867	550.7	16.64	12.61	.2754	83.47	87.41	114.77
35	0.004907	34.35	14.35	0.7888	0.2422	5.206	5.285	13.69	115	0.08123	568.6	16.67	13.02	0.2766	86.18	90.29	117.89
36	.005109	35.77	14.38	.8213	.2423	5.421	5.506	14.15	116	.08390	587.3	16.70	13.44	.2778	89.01	93.29	121.13
37	.005319	37.23	14.41	.8545	.2424	5.643	5.733	14.61	117	.08663	606.4	16.73	13.88	.2790	91.92	96.37	124.45
38	.005534	38.74	14.44	.8894	.2425	5.872	5.968	15.09	118	.08944	626.1	16.76	14.33	.2803	94.90	99.53	127.85
39	.005757	40.30	14.47	.9251	.2426	6.108	6.211	15.57	119	.09233	646.3	16.79	14.80	.2816	97.96	102.78	131.34

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(Tabulations for 22 in., 23 in., 24 in. and 25 in. were published in July, August, September and October.)

Properties of Mixtures of Air and Saturated Water Vapor—(Continued)

Barometric Pressure 26 In. of Mercury

All tabulated values (except temperatures and pressures) are quantities per pound of dry air in the mixture.

Temp Deg Fahr	Absolute Humidity		Volume		Humid Specific Heat	1061w	Enthalpy (Heat Content)	
	Pounds	Grains	Dry Air Cu Ft	Volume Increase of Humid Air Per Cent			Water Vapor Btu	Air-Vapor Mixture Btu
t	w	w	v_D		s''		h_V	h
1	2	3	4	5	6	7	8	9
120	0.09537	667.6	16.82	15.28	0.2829	101.18	106.21	135.01
121	.09848	698.4	16.84	15.77	.2843	104.49	109.72	138.76
122	.1017	711.6	16.87	16.29	.2858	107.86	113.30	142.58
123	.1050	734.9	16.90	16.82	.2873	111.39	117.05	146.57
124	.1084	758.9	16.93	17.37	.2888	115.03	120.92	150.68
125	0.1120	783.8	16.96	17.93	0.2904	118.80	124.94	154.94
126	.1156	809.5	16.99	18.52	.2920	122.69	129.08	159.32
127	.1194	836.1	17.02	19.12	.2938	126.73	133.37	163.85
128	.1234	863.5	17.05	19.76	.2955	130.88	137.79	168.51
129	.1274	891.9	17.08	20.41	.2973	135.18	142.37	173.33
130	0.1317	921.8	17.11	21.08	0.2993	139.72	147.22	178.42
131	.1360	952.1	17.13	21.77	.3012	144.32	152.11	183.55
132	.1404	982.9	17.16	22.48	.3032	148.98	157.08	188.76
133	.1451	1015.7	17.19	23.23	.3052	153.95	162.38	194.30
134	.1499	1049.6	17.22	23.99	.3075	159.09	167.86	200.02
135	0.1549	1084.3	17.25	24.80	0.3097	164.35	173.47	205.87
136	.1601	1120.4	17.28	25.61	.3120	169.81	179.30	211.94
137	.1654	1158.1	17.31	26.47	.3145	175.53	185.43	218.31
138	.1711	1197.5	17.34	27.37	.3170	181.51	191.80	224.92
139	.1768	1237.5	17.37	28.28	.3196	187.57	198.29	231.65
140	0.1828	1279.3	17.40	29.23	0.3222	193.90	205.05	238.65
141	.1889	1322.2	17.42	30.22	.3250	200.41	212.01	245.85
142	.1955	1368.2	17.45	31.25	.3280	207.37	219.45	253.53
143	.2020	1414.2	17.48	32.32	.3309	214.35	226.92	261.25
144	.2089	1462.6	17.51	33.42	.3340	221.69	234.77	269.33
145	0.2162	1513.2	17.54	34.58	0.3373	229.36	243.00	277.80
146	.2238	1566.4	17.57	35.77	.3407	237.42	251.63	286.67
147	.2316	1621.1	17.60	37.01	.3442	245.71	260.50	295.78
148	.2397	1678.0	17.63	38.31	.3479	254.33	269.75	305.27
149	.2481	1736.8	17.66	39.66	.3517	263.26	279.31	315.07
150	0.2570	1799.1	17.69	41.07	0.3557	272.70	289.43	325.43
151	.2662	1863.5	17.71	42.54	.3598	282.46	299.90	336.14
152	.2755	1928.4	17.74	44.06	.3640	292.29	310.45	346.93
153	.2858	2000.7	17.77	45.66	.3686	303.24	322.19	358.91
154	.2962	2073.7	17.80	47.31	.3733	314.31	334.07	371.03
155	0.3072	2150.6	17.83	49.07	0.3783	325.97	346.59	383.79
156	.3186	2230.3	17.86	50.88	.3834	338.06	359.60	397.04
157	.3303	2312.4	17.89	52.77	.3887	350.49	372.95	410.63
158	.3430	2401.1	17.92	54.76	.3944	363.93	387.40	425.32
159	.3561	2492.4	17.95	56.84	.4002	377.78	402.28	440.44

Temp Deg Fahr	Absolute Humidity		Volume		Humid Specific Heat	1061w	Enthalpy (Heat Content)	
	Pounds	Grains	Dry Air Cu Ft	Volume Increase of Humid Air Per Cent			Water Vapor Btu	Air-Vapor Mixture Btu
t	w	w	v_D		s''		h_V	h
1	2	3	4	5	6	7	8	9
160	0.3698	2588.7	17.98	59.03	0.4064	392.38	417.97	456.[illegible]
161	.3842	2689.5	18.00	61.32	.4129	407.66	434.40	475.[illegible]
162	.3994	2795.9	18.03	63.74	.4197	423.78	451.74	496.[illegible]
163	.4153	2907.1	18.06	66.27	.4269	440.63	469.87	518.[illegible]
164	.4321	3024.7	18.09	68.95	.4345	458.46	489.05	541.[illegible]
165	0.4498	3148.3	18.12	71.75	0.4424	477.18	509.21	565.[illegible]
166	.4685	3279.4	18.15	74.73	.4508	497.07	530.61	590.[illegible]
167	.4881	3416.8	18.18	77.85	.4597	517.88	553.03	616.[illegible]
168	.5091	3563.6	18.21	81.18	.4691	540.13	576.99	643.[illegible]
169	.5311	3717.4	18.24	84.67	.4790	563.44	602.10	672.[illegible]
170	0.5545	3881.4	18.27	88.40	0.4895	588.31	628.90	699.[illegible]
171	.5793	4054.8	18.29	92.33	.5007	614.58	657.22	728.[illegible]
172	.6057	4240.2	18.32	96.54	.5126	642.69	687.52	760.[illegible]
173	.6341	4438.6	18.35	101.1	.5253	672.77	719.95	794.[illegible]
174	.6642	4649.3	18.38	105.8	.5389	704.71	754.39	830.[illegible]
175	0.6960	4872.1	18.41	110.9	0.5532	738.47	790.81	868.[illegible]
176	.7304	5112.8	18.44	116.3	.5687	774.95	830.17	909.[illegible]
177	.7674	5371.7	18.47	122.2	.5853	814.20	872.52	953.[illegible]
178	.8070	5648.9	18.50	128.5	.6032	856.22	917.87	999.[illegible]
179	.8499	5949.6	18.53	135.3	.6225	901.79	966.98	1049.[illegible]
180	0.8958	6270.5	18.56	142.8	0.6431	950.42	1019.5	1102.[illegible]
181	0.9471	6629.4	18.58	150.8	.6662	1004.8	1078.2	1161.[illegible]
182	1.001	7009.1	18.61	159.3	.6906	1062.4	1140.4	1184.[illegible]
183	1.063	7437.5	18.64	169.1	.7181	1127.3	1210.5	1254.[illegible]
184	1.129	7900.2	18.67	179.6	.7479	1197.4	1286.3	1330.[illegible]
185	1.201	8409.1	18.70	191.1	0.7806	1274.6	1369.6	1414.[illegible]
186	1.282	8971.9	18.73	203.8	.8168	1359.9	1461.8	1506.[illegible]
187	1.371	9597.7	18.76	218.0	.8570	1454.7	1564.3	1609.[illegible]
188	1.471	10,300	18.79	233.9	.9021	1561.2	1679.3	1724.[illegible]
189	1.584	11,091	18.82	251.9	.9530	1681.0	1808.9	1854.[illegible]
190	1.712	11,987	18.85	272.2	1.011	1816.9	1955.6	2001.[illegible]
191	1.859	13,013	18.88	295.4	1.077	1972.4	2123.7	2169.[illegible]
192	2.028	14,197	18.90	322.3	1.153	2151.8	2317.7	2363.[illegible]
193	2.226	15,584	18.93	353.7	1.242	2362.1	2545.1	2591.[illegible]
194	2.460	17,222	18.96	390.9	1.347	2610.4	2813.6	2860.[illegible]
195	2.741	19,189	18.99	435.4	1.474	2908.5	3136.1	3182.[illegible]
196	3.085	21,596	19.02	490.0	1.628	3273.3	3530.6	3577.[illegible]
197	3.516	24,613	19.05	558.2	1.822	3730.6	4024.9	4072.[illegible]
198	4.070	28,490	19.08	646.1	2.072	4318.3	4660.6	4708.[illegible]
199	4.627	32,390	19.11	734.4	2.322	4909.5	5300.5	5348.[illegible]
200	5.839	40,872	19.14	926.5	2.868	6195.1	6690.8	6738.[illegible]

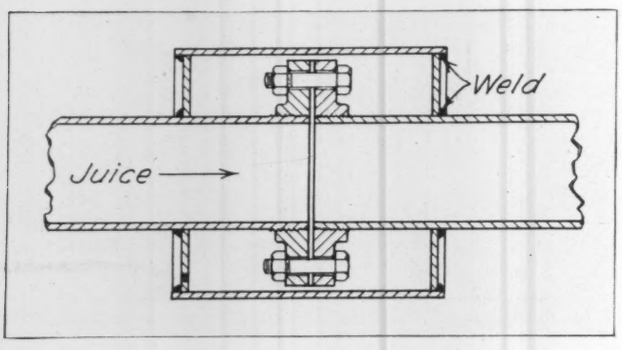
How a Leaky Pipe Flange Was Repaired on the Run

A pipe line carrying sugar juice sprung a leak in the threads of a flange.

It would have meant a shutdown of the factory to replace the old flange with a new one. The electric welder operator was of the opinion that he could make a temporary repair that would last until a permanent repair could be made without shutting down the factory.

He first cut from a boiler plate two disc washers larger in diameter than the flange, the opening in each being the size of the pipe. These washers were cut in half and placed on the pipe, and the halves welded together.

Sketch showing how welder repaired leaking pipe flange



The washers were then welded to the pipe. The lower one of the washers had a hole drilled at the lowest point so the trapped juice could be drained out during the welding operation.

A pipe large enough to slip over the flange and of sufficient length to reach to the washers was cut in halves and placed over the flange and welded together and to the washers. As soon as the welding was completed a plug was screwed into the opening left in the lower washer and the juice leak was completely stopped.

This job cost very little, as all material was scrap, and it saved a shutdown.—A. J. Forbess.*

Trend to Heavy Oils

There is a new trend developing in the burning of medium heavy oils in very small buildings. A number of burners are on the market for burning No. 5 oil in medium and small houses. This has been brought about in many cities by a price differential of 3c per gal between No. 3 oil, which has long been used as an automatic fuel, and No. 5 oil. For homes using 3000 gal or more per year, this represents a saving of $90 which of course makes it possible for the owner to consider spending two or three hundred dollars more for an oil burner of this type.—Paul R. Unger.†

*Chief Engineer, Amalgamated Sugar Co.
†Mid-West Heat Service Co., Combustion Engineers.

The Importance and Attainment of Smokeless Boiler Plant Operation*

By J. F. Barkley†

THE importance of the prevention of smoke from a community standpoint is never questioned. In this connection, the word "smoke" is used to mean the visible unburned combustibles coming from a fuel burning furnace. Everyone is ready to agree that the absence of such particles from the air is quite desirable for better health and sanitary conditions. Various proofs and statements have been made to show that smoke prevention is also an economic procedure from the community standpoint.

Smoke continues to be produced, however, because its elimination depends upon the summation of the actions of many individual plants; and the individual plant owner frequently has a few more and somewhat different thoughts on the matter when considering his own plant. His plant is only one of many; the situation is such that it does not appear practical to prevent smoke entirely; he cannot keep watching the firemen continually, and he has trouble getting good firemen; a change in the fuel burning equipment would cost too much; there are so many other things requiring attention; his plant is a somewhat special case; he is making no more smoke than some others; improvement in his plant would not help the community much. When such reasoning is applied to each individual case, it is apparent that the community as a whole will have considerable smoke.

There are a number of reasons why the individual plant owner from his own standpoint should not produce smoke. Smokeless combustion gives the following results:

a) Improved combustion efficiency. Smoke is unburned fuel; it frequently is accompanied by invisible unburned gases. With the lower furnace temperatures—so often found in smaller boilers—an appreciably greater amount of fuel must be used when smoke is produced. In general, the cost of the elimination of smoke with improved combustion efficiency will be covered amply by the decreased cost of fuel. The usual operating engineer likes to use as little fuel as possible to accomplish the needed results. The total fuel bill, however, is an expenditure that is rather difficult to analyze and judge. Many owners or superintendents will pay a fuel bill without much question or with only a few repeated cautions; but if the operating engineer wishes to spend a little money for an instrument, or for some device or construction that will lower the fuel bill somewhat, or if he should happen to have increased stoker and furnace maintenance because of carrying better furnace temperatures and combustion efficiencies, he is apt to be open to considerable questioning and perhaps criticism. He may decide that it is simpler to bury the matter in the fuel bill.

b) Improved boiler conditions. The production of smoke coats the boiler heating surfaces with soot, lowering its efficiency in taking the heat from the gases and thereby raising the fuel bill. The draft loss through the boiler increases. There is also increased cost or labor for cleaning the boiler heating surfaces.

c) Improved breeching and stack conditions. Soot from the smoke may accumulate in the boiler breeching and stack, thereby lowering the draft to a point where it is difficult to carry the boiler load. Such soot has been known even to catch fire, which might result in more serious difficulties.

d) Cleaner immediate surroundings. Although the smoke from the stack of a building usually makes a neighboring building dirty, it may also to some extent make the building having the stack dirty, thus increasing cleaning costs. Elimination of the smoke improves the good will of the neighborhood toward the owner.

e) Better neighborhood tone. The character of the immediate neighborhood is improved by the elimination of smoke. This tends toward better building income.

f) Freedom from troubles with the smoke inspector. Where there are smoke regulations, smoke production causes at least intermittent trouble with inspectors, possibly leading to convictions and fines. This may be one of the strongest arguments that can be presented to the individual mind.

The Cause of Smoke

Whenever a fuel not inherently smokeless is burned, smoke may readily be produced; it is necessary to take care of all factors that may be helpful in lowering smoke emission. It is desirable for each fireman to understand the cause of smoke and to have a few fundamental ideas about combustion. When a fuel burns it is uniting with the oxygen of the air. To unite with this oxygen, that forms about 20 per cent of the air, the fuel must be hot. Therefore, for anything to burn it must be kept hot enough, it must have enough air supplied to it, and as a practical matter it must have enough room in which to burn. When a piece of coke burns the oxygen unites with it at its surfaces, its hotness being maintained by the heat of the burning. When a piece of coal burns, the action is a little different. As it heats it gives off some smoky gas called volatile matter. A piece of paper when warmed will give off considerable volatile matter before it bursts into flame. Smoke from a stack is

The question of preventing smoke is approached here from the standpoint of the operator of the individual boiler plant. Mr. Barkley tells why the individual plant owner should not—for his own advantage — produce smoke, explains in non-technical terms how smoke is formed, and says that the chief cause of smoke production in the ordinary plant is the way it is operated. He then gives some operating pointers, and cites the value of smoke indicating devices and the application of steam-air jets as practical helps

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†Supervising Engineer, Fuel Economy Service, Bureau of Mines, Department of the Interior.

caused by such volatile matter leaving the furnace unburned. It did not unite with oxygen for such causes as never being hot enough, cooling too quickly, or never being mixed with oxygen. It might not have gotten hot enough because too much was given off quickly; it might have gotten hot but was cooled down immediately on account of striking some cold surfaces of the boiler; it might not have been mixed with air owing to insufficient draft, lack of air openings, or lack of turbulence. With such fundamental thinking, it is possible for one to watch a smoke producing fire and soon decide what is the difficulty.

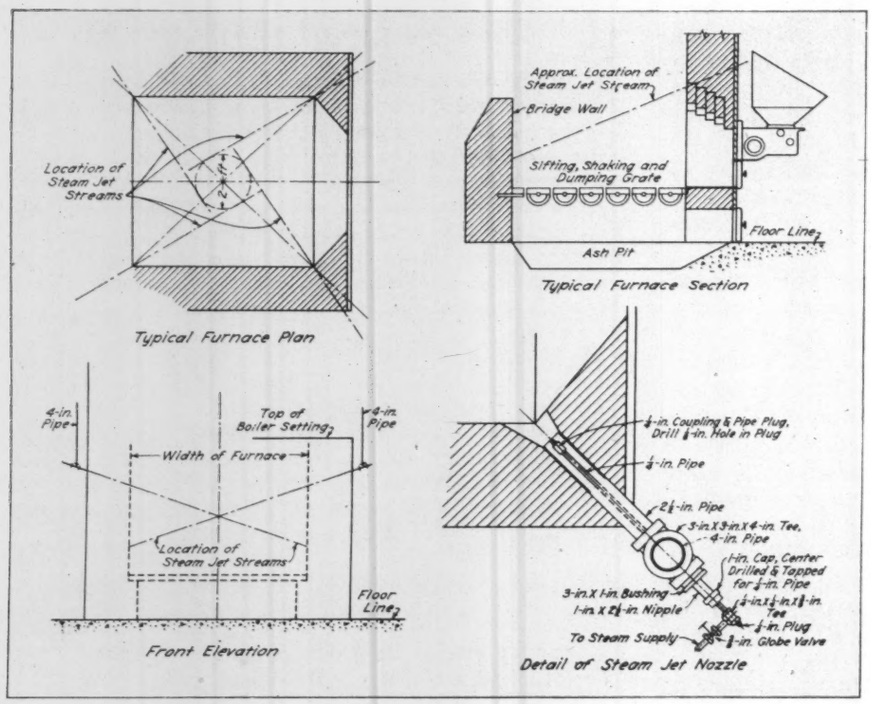


Diagram showing use of steam-air jets

The fuel burning equipment must be at least reasonably well adapted to the fuel being burned. It should provide for a feeding of the fuel that does not cause too much smoky gas to be given off at one time. Enough space must be provided for the flames to have ample room to develop. Ample overfire draft should be available. The equipment should be kept in good condition, especially as to undesirable air leakage. Needed changes in furnace arrangement sometimes may be made at low cost. Where very much furnace change is necessary, it is usually best to have an experienced man on such problems. Help is usually given gladly by city smoke abatement engineers.

Eliminating Smoky Operation

Probably the chief cause of smoke production in the ordinary plant is the way it is operated. However well planned the equipment, much smoke may be produced because of faulty operation. Many buildings have boiler rooms so situated that the fireman has no idea what is happening at the stack outlet. One of the biggest helps in eliminating smoky operation is a means for the fireman to know at all times exactly how much smoke is being emitted from the stack. There are many devices for this purpose. A simple one is an ordinary mirror so placed that the top of the stack can be seen. Periscopes of various types may be used. There are available on the market instruments that give a daily record of the smoke emission, throw colored lights on and off, and ring bells. One very effective scheme, from the standpoint of getting quick action from the operator, consists of the blowing of a raucous horn when the smoke density passes a chosen point.

Smoke indicating devices are also quite helpful to a fireman in teaching him what produces smoke. Many boilers are equipped with the underfeed type of stoker. On this stoker, the coal is pushed upward toward the burning fuel bed above. The coal is gradually heated as it approaches the hot fuel bed and slowly gives off its volatile matter, which is readily ignited by the hot fuel bed and burned. By the time the coal reaches the fuel bed it has become mostly burning coke. This fuel burning principle is ideal and can be applied quite effectively, but as a practical matter considerable smoke may be produced because of undesirable operating conditions. Manhandling the fuel bed, thereby stirring up green coal into the hot zones, usually causes a great release of volatile matter which does not burn. Such operations as feeding the coal too rapidly or shoveling in some extra coal by hand to bring up the load quickly; cleaning the fires rather carelessly, disturbing the fuel bed unduly; improper adjustment of the air supplying auxiliaries; poor draft regulation; and stopping the stoker and air supply with fuel bed conditions that result in much warmed coal continuing to give off volatile matter—all result in smoke. Sometimes considerable smoke may be produced by a poor choice of coal. In general, these causes also apply to other types of equipment. Where soft coal is hand fired much skill is required to keep down smoke.

Use of Steam-Air Jets

When undesirable smoke is produced under reasonable conditions of operation of the equipment available, its satisfactory elimination frequently can be obtained by means of steam-air jets. This device injects air into the furnace by means of a small jet of steam. For good results in the larger furnaces an available steam pressure of at least about 50 lb ga is desirable. Sometimes lower pressures may be used, or even air jets alone, but in general the higher pressures are needed to effect good mixing and reach fully across the furnace. As illustrated in the figure, the jets may be placed at the

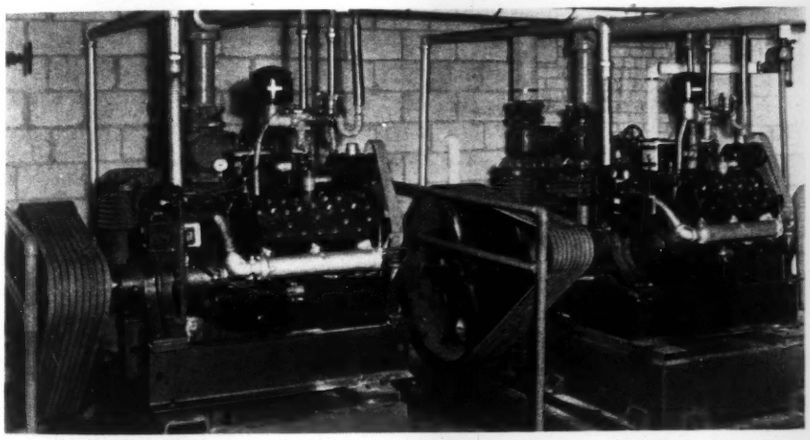
four corners of a furnace. They are tilted downward and create a circular turbulence in the center of the furnace, each jet spinning the gases in the same circular direction. They may also be arranged so that two of the jets—one on each side of the furnace—tend to spin the gases in the opposite direction to the other two jets. For many furnaces, one jet at each front corner is ample. The jets may be simply made in various ways from ordinary pipe and fittings, the detail in the figure showing one of many methods that has been used in forming them. The small steam nozzle had best be placed a foot or more back of the air pipe outlet into the furnace.

Objections have been raised because of their use of steam and their noise. It is seldom necessary to use them continuously, only turning them on at critical times. The amount of steam used can readily be calculated. Such calculations show, as an example, that at 100 lb steam pressure on the nozzles, two ⅛ in. jets, used continuously, would take about 1.6 per cent of the steam produced by a 300 hp boiler running at rating. The improvement in the combustion, however, ordinarily more than makes up for this loss. If the air inlets to the jets are open to the air at the boiler front, much noise is produced and the operators are not apt to use them. If the air is piped up from the stoker windbox or taken from air channels in the settings, the noise nuisance disappears. The proper application of steam-air jets has been the big practical help in many smoke abatement problems.

Automobile Engines Running on Natural Gas Drive Compressors for Conditioning Theater

By Harry W. Smith, Jr.*

OWNERS of the Beverly theater, Detroit, Mich., have achieved operating economies by using adapted automobile engines run on regular city natural gas to power the two 60 ton compressors which furnish refrigeration for air conditioning their 1487 seat theater. The set-up consists of two V8 engines equipped with high altitude or high compression heads (7½ to 1 ratio) and natural gas carburetors. The engines are



Two automobile engines converted to natural gas operation drive the compressors for air conditioning Detroit's Beverly theater

fitted with governors which limit the top speed to 2200 rpm and are connected by V belts to the two compressors. Although the engines are listed as 85 hp models, they actually in this case deliver but 54 hp at top speed, adapted as they are to continuous operation on natural gas. Modulators are provided so that either engine may be slowed down as gradually as required to 1100 rpm. Thus, there being two engines and two compressors, any degree of adjustment from one-fourth capacity to full capacity operation is obtainable.

The record of operation during the first full season (June 15 to Labor Day, 1938) is "proof of the pudding." Between these dates the air conditioning system was in daily operation from 12:00 noon to 12:30 a. m., Sundays and holidays included, and on occasional hot days during the few weeks both preceding and succeeding this period it was used as required. The coils connected to the compressors were chilled to 30 F, and a dry bulb temperature of 74 F was maintained in the theater auditorium itself. Over the entire season the cost of operating the gas engines was $331.82, broken down as follows: Cost of gas fuel (five monthly bills) $132.24; cost of water for cooler, $176.08; oil, $3.90; cleaning filters, $19.60.

Maintenance and care of the engines has been insignificant, according to the owners. No attention other than starting and stopping them, changing oil once during the season, and having oil filters cleaned, has been required. Batteries are traded in for new ones each year, substantial allowances upon the old ones being obtained. Thus, at a slight premium, ignition worries are nullified. A non-paraffin-base oil is used; this is important because in continuous operation on natural gas the crankcase temperature is 30 to 40 F higher than that which would be attained in regular automotive service for the same engine, and therefore the oil must contain a minimum of unstable hydrocarbons which can break down into gums at elevated temperatures.

Perspiration from worker's brows not only slowed up making watch parts but also made necessary an expensive drying process through which the pieces had to be run before assembling at the Bulova Watch Co., Woodside, L. I., N. Y.

A 65 ton central station conditioning system maintaining the temperature at 80 deg and relative humidity at 50 per cent was installed to solve the problem.

*American Gas Association.

Lease Forms for Air Conditioning

THE manager of a newly air conditioned building recently requested information regarding an air conditioning clause to be used in lease forms. In order to prepare a reply to his question, managers of a number of air conditioned buildings were asked for copies of the clauses they used.

A New Orleans building uses the following clause:

In the event air conditioning is installed in the ..... Building, rental under this lease shall be increased by $..... from the date such air conditioning shall become effective in the premises hereby leased.

This clause, the manager states, was inserted in all leases prior to the installation and operation of the air conditioning system and allowed the building to increase the rental when the air conditioning became effective.

A building manager in the middle west forwarded the following "Amendment to Lease" form which was used at the time air conditioning was installed:

WHEREAS, the .....Company, as lessor, and ....., as lessee, entered into a written lease dated the ..... day of ....., 19...., covering certain space in the ..... Building; and

WHEREAS, the said lessor proposes to furnish air conditioning in said building commencing with June 1, 1937;

NOW, THEREFORE, in consideration of the lessor's furnishing such air conditioning, the lessee agrees to pay, in addition to the monthly rent provided in said lease, a further sum of $..... per month, to be paid with the rental provided for in said original lease, commencing on June 1, 1937.

Lessor shall not be liable for any damages for failure to furnish air conditioning, or for any stoppage of said service not directly due to a lack of ordinary care by lessor or arising from riot, strikes, unavoidable accident, or casualty, or other like or unlike causes beyond the control of the lessor, or for any stoppage of said service for needful repairs or improvements, provided lessor uses reasonable diligence to resume such service.

If for any reason it is impossible to commence furnishing air conditioning on the first of June, 1937, the additional payment provided for herein shall be reduced proportionately.

If the lessor decides not to furnish air conditioning in said building and notifies the lessee accordingly prior to June 1, 1937, this amendment to lease shall become null and void.

A Texas building manager replied that they do not have a lease form which incorporates a clause relating to air conditioning, because as most leases provide that the lessor shall heat the premises when necessary for comfort, they feel that air conditioning might be handled in the same manner, and the expense of the service be set up in the rental schedule in the regular way. Having few leases, this building merely advanced the rental rate and had no difficulty whatever in putting the new rates into effect. When installation of air conditioning was started they advised their tenants that there would be an increased charge for the service not to exceed 20 per cent of existing rentals. Upon completion of the installation and an estimate of the cost, the increase was held to about 16⅔ per cent.

The manager of this building states that were they to insert an air conditioning clause they would like to have it read something as follows:

Lessor does hereby obligate itself to maintain, furnish, and provide to the lessee complete air conditioning service, reasonably sufficient properly and adequately to air condition the premises herein demised, continuously during the usual and ordinary business hours kept and observed by lessee, it being understood that lessor shall not be liable for interruption in service of reasonable duration occasioned by the necessity of repairs to machinery or equipment, and the failure to furnish such air conditioning shall not operate as an eviction of the lessee nor shall lessee be entitled to an abatement of rental for such period of temporary interruption in the said services.

Of course, he points out, if summer conditioning only was furnished, such a clause should be limited to the normal summer season, such as May 1 to October 1, or limited to a certain number of maximum hours, subject to climatic conditions and the desire of the parties involved.

The last reply to the question came from a California building manager who advised that they have not found it necessary to add any special clauses to their leases as a result of the installation of air conditioning in their buildings, and that nothing had transpired since the installation to cause them to think that any special clauses relative to air conditioning are necessary.

Install a Booster Pump on Hot Water Heating System?

A Pennsylvania reader of HPAC asks: "Would you recommend the installation of a booster pump and an additional section to the boiler on a present gravity hot water heating system in order to raise the heating capacity of the whole system?"

ANSWER (BY SAMUEL R. LEWIS*)—If the gravity hot water plant in general is all right, and if the boiler already is not too long, the proposed changes certainly would permit an increase in the amount of coal burned with corresponding increase in the amount of heat delivered.

This seems to be a specific answer to the question. In my own house I added a booster pump to an excellent gravity hot water heating system, not to burn more coal and deliver more heat, but to burn less coal more efficiently, heat the domestic hot water as well, and to prevent the tendency for frequent overheating, which existed with gravity circulation. After five years experience I know I was wise.

We have recommended the installation of a great many booster pumps on old gravity systems for the above reasons, as well as to quicken the results following an increase in combustion rate and to reduce the tendency, with sluggish gravity systems, for overheating the room before the radiator can cool down following a demand for more heat.

I can think of no reason why the solution suggested should not be followed, unless there is something wrong with the boiler, piping, radiators, etc. With increased firing the stoker controls particularly should be considered, to maintain ignition and yet not over-run. There should be intelligent observation to whether the local heat transmitters are radiators, convectors or unit heaters.

*Consulting Mechanical Engineer. Member of Board of Consulting and Contributing Editors.

Heating · Piping and Air Conditioning

AMERICAN SOCIETY of HEATING and VENTILATING ENGINEERS

Journal Section

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BULLETIN BOARD

Every Society member has a part in the Research program and can benefit from the data that are developed at the ASHVE Research Laboratory and in cooperating institutions. Much of the program is financed by an allotment from membership dues. There are 23 projects under way and reports on many of them will be made at the 46th Annual Meeting in Cleveland, Ohio, January 22-25, 1940.

• • •

Cleveland will be the mecca of the heating, ventilating and air conditioning industry. Attention will be focused on events during the week of January 22nd because of the ASHVE Meeting, the 6th International Heating and Ventilating Exposition at Lakeside Hall, and the mid-winter meeting of the *National Warm Air Heating and Air Conditioning Association*. Make plans early and Be On Hand to Meet Friends from everywhere in Cleveland.

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A Laboratory for Use in Determining the Influence of Various Temperatures and Relative Humidities on the Final Quality of Air-Cured Tobacco†

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THE Kentucky Agricultural Experiment Station recently completed an air conditioned laboratory for research in the curing of tobacco. The laboratory consists of nine chambers, in which the tobacco is housed during the process of curing, together with the necessary equipment for maintaining within each chamber any desired temperature and relative humidity. The laboratory was designed for experiments with air-cured tobacco, and the chief objective was the determination of the conditions required for obtaining cured tobacco of the best quality.

For the purpose of design the conditions specified for the chambers ranged from 50 F to 150 F, and from 25 to 100 per cent relative humidity. In the beginning the planning followed two criteria. First, it was assumed that, with all chambers in use at the same time, any chamber might be operated at any combination of temperature and humidity between the upper and lower extremes; and second, that all nine chambers might be operated simultaneously at either extreme. These assumptions lead to the tentative selection of equipment which appeared to be unreasonably large for the general needs of the project. Later, the assumptions were modified, and the planning followed a course in which it was assumed that equipment would be installed to operate nine chambers simultaneously over a relatively narrow range of conditions compared to the original specifications; and that the extreme conditions, when required for special experiments, would be obtained by using the full capacity of the equipment in a limited number of chambers. From the designer's point of view, the outstanding feature of the laboratory is the small size and number of principal units it contains, compared with its flexibility, or adaptability to a wide range of conditions.

Curing Chambers

The curing chambers are contained in two separate cabinets. One cabinet contains three two-tier chambers. The second cabinet has the same over-all dimensions as the first one but is divided horizontally between upper and lower tiers so that it contains six chambers instead of three. The tobacco is placed upon sticks about 4 ft long which rest upon the tier rails; and the rails are supported by chains suspended from a framework borne by platform scales located on top of the chambers. There are 12 scales—one for each tier, or pair of rails, so that the weight of each lot of tobacco can be obtained separately at any time.

Each chamber is 5 ft square in inside plan dimensions. The smaller chambers are about 5 ft high; and the larger two-tier chambers are about 10 ft high. With spacings of six plants per stick and six sticks per pair of rails, each of the smaller chambers will hold 36 plants, while the larger chambers will each hold 72 plants.

In a normal season an average plant will weigh about 4 lb immediately after cutting, so that the normal charge for each chamber is 144 lb for the smaller chambers and 288 lb for the larger ones. During the process of curing, which may extend over a period of from two to four weeks, depending upon the conditions, the tobacco will lose by drying about 90 per cent of its initial weight. It is obvious, then, that, from the point of view of the selecting of air conditioning equipment for the laboratory, the problem of dehumidifying was the chief one to be considered; and furthermore, that the most important physical property of the tobacco affecting the design was the rate of loss of water by evaporation at various temperatures and relative humidities during the first few hours after the tobacco was placed in the chambers. The problem of heating and humidifying was not considered as offering any difficulty, either from the point of view of the method to be employed or the selecting of equipment.

Air Conditioning System

Several combinations of different kinds of equipment were considered, including both unit systems in which each chamber was to be equipped independently, and central systems for supplying air to all chambers. Finally, a central system was chosen which embraced a silica-gel dehumidifier as a source of dry air and a condensing unit and evaporator as the chief means of cooling. The system is shown diagrammatically in Fig. 1.

The right-hand side of the diagram shows the equipment duplicated for each chamber, and the left-hand side shows the central system. Looking first at the chamber system, the major units are a fan (F), an

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electric heater (EH), and a humidifier (HM). Between the chamber outlet and the fan there are a number of branches from the main duct which either extend to the central system or open into the room. To control the flow of air in these branch ducts there are five dampers designated 1, 2, 3, 4 and 5. These dampers may be adjusted either manually or through the automatic control system. Dampers 1, 2 and 5 are double mixing dampers. Dampers 3 and 4 are single dampers independently operated.

Depending upon the positions of the dampers, the chamber can be operated in any one of the following three ways:

1. By closing dampers 1, 3 and 4 the chamber can be cut off from the room and from the central system, and the regulation of temperature and humidity obtained by means of the chamber heater and humidifier.

2. By closing damper 3 and opening damper 5 to the room, the chamber can be operated on all-room air, or on any mixture of room air and recirculated air. For this method of operation, dampers 1 and 4 operate together so that when room air is drawn into the system through 5 and 1 an equal amount is exhausted through 4. In addition to regulating the heater and humidifier, the proper mixing of room air and recirculated air may be used to assist in controlling the temperature and humidity.

3. By closing dampers 4 and 5 to the room, the chamber may draw all of its air from the central system, or it may operate on a mixture of central system air and recirculated air, using this mixture alone or in conjunction with the heater and humidifier to regulate the temperature and humidity. When this method is used, dampers 1 and 3 operate together so that when air is drawn from the central system an equal amount is returned to the central system.

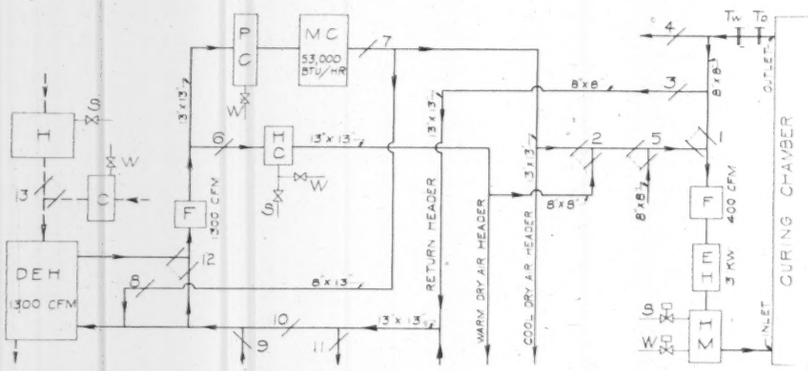

Fig. 1—Diagram of the air conditioning system

While these three methods are basic, it is obvious that several variations are possible. For example, damper 5 may be set in an intermediate position so as to mix room air with air from the central system. By closing damper 3 and operating 4 and 1 together, at the same time air is drawn from the central system the excess can be exhausted to the room instead of returning it to the central system. Also, when operating on room air, the excess may be delivered to the central system through 3 instead of returning it to the room through damper 4.

The particular method employed for a given chamber depends upon the conditions maintained within the chamber in comparison with other chambers, the temperature and humidity of the room air, and the quality of the air obtainable from the central system.

The major equipment of the central system consists of a main cooler connected to a refrigerant condensing unit, a silica-gel dehumidifier, and a fan. Beginning with the central fan (F) shown on the left-hand side of the diagram, and tracing the flow of air through the system, it will be noted that the duct leaving the fan divides into two branches. The main cooler (MC) is located in one of these branches and a fin-type coil heater (HC) is located in the other branch. For reasons to be explained later the coil is also connected to the water supply so that it can be used sometimes as a cooler.

The duct leaving the main cooler terminates in a header from which separate branches lead to each chamber. There is also a connection to this header which leads back to the central system. Likewise, the duct leading from the heating coil terminates in a header to which individual chamber branches are connected. These two headers, respectively, supply cool-dry air and warm-dry air to the chambers. At the chamber the separate branches are connected together through a mixing damper, designated 2 in the diagram. This damper is adjusted automatically to mix the cool air and the warm air from the central system and thus to supply the correct mixture to maintain the desired temperature within the chamber. The damper may be used as the sole means of regulating the temperature, or it may be operated in conjunction with the electric heater installed within the supply duct of the chamber. The mixing of the dry air from the central system with return air from the chambers by automatic positioning of damper 1 is the primary means of regulating the humidity, although for certain conditions this method may be supplemented by regulation of the humidifier located near the inlet of the chamber.

The branch ducts which return air from the chambers to the central system are connected to a header which in turn connects to the inlet of the silica-gel dehumidifier. The discharge from the dehumidifier is connected to the suction side of the main circulating fan. There is a by-pass around the dehumidifier by which the return air from the chambers may be conveyed directly to the inlet of the main fan. This by-pass is opened or closed by hand adjustment of the mixing damper 12. Moreover, this damper may be set in any intermediate position so as to proportion the relative amounts of dehumidified and untreated chamber air returned to the inlet of the central fan.

Ahead of the dehumidifier and the central fan there are three dampers designated as 9, 10 and 11. Damper 10 is in the main return duct; damper 11 is in an exhaust duct connected to the main return; and damper 9 is in a branch duct which may be used to supply room air to the central system. These three dampers are linked together and may be adjusted by hand through a motor and graduated switch so as to either return all of the chamber air to the central system, to mix return air from the chambers with room air, or to exhaust the chamber air and supply all room air to the central system. Which one of these methods is employed depends

upon the relative quality of the mixture of return air from the chambers compared with the quality of the room air. For example, if the mixture of return air from the chambers is at a low temperature and low absolute humidity, it would be desirable to return all of the air to the central system in order to conserve refrigeration and to lessen the load on the dehumidifier. On the other hand, if the mixture of return air is at a relatively high temperature and humidity, better operation may be obtained from the standpoint of both economy of operation and regulation if all of the chamber air is exhausted and the supply taken into the central system directly from the room; or, because of the operating conditions in a given chamber, dampers 3 and 4 may be set to exhaust the chamber air directly, and the difference made up by supplying air to the central system through damper 9. Since dampers 9, 10 and 11 operate together, any tendency for the pressure in the central system to change appreciably from normal is corrected by the relative amounts of air exhausted to or supplied from the room.

While the exhaust ducts have been described as exhausting into the room, actually, all of the exhaust branches were connected into one pipe which ran outside of the building.

The characteristics of the dehumidifier are such that it is more efficient when called upon to treat air at a low temperature. With this point in view, the cross-connection between the cool air at header and the main return was installed so that whenever there was both an excess of cooling capacity and an excess of air supply some of the excess air could be recirculated, or returned to the dehumidifier, for the purpose of reducing the mean temperature of the entering air and thereby increase the efficiency of moisture removal. Damper 8 was installed in the cross-connections for the purpose of regulating the amount of cool air returned to the dehumidifier.

The diagram shows a precooler (PC) ahead of the main cooler. The precooler is supplied with water from the city mains, and its purpose is chiefly to cool the air leaving the dehumidifier to approximately water temperature before it enters the main cooler. The precooler can be used, of course, without the dehumidifier.

For some test conditions the temperature of the air leaving the dehumidifier may be higher than that desired in the warm-dry air duct. To reduce the temperature the coil in the warm air duct may be used as a cooler by supplying it with water from the city mains.

From the foregoing description it is obvious that the central system may be operated in a number of different ways. The use of any particular combination will depend, of course, upon the extreme conditions selected for the tests in the curing chambers. In general, the lowest dew-point temperature maintained within any chamber will determine the maximum allowable dew-point of the dry air in the cool and warm air ducts of the central system. The lowest dry-bulb temperature maintained within any chamber will determine the maximum allowable temperature of the cool air supply; and the highest temperature within any chamber will determine the minimum temperature of the warm air supply. Thus, in principle, the central system is adjusted to meet the widest range of conditions prescribed for a given series of simultaneous tests, and the intermediate conditions required for some chambers are obtained by means of the mixing dampers.

For some test conditions it may be desirable to regulate either the individual chamber heaters or humidifiers, or both, to obtain close regulation of the temperature and humidity within the chambers. In fact, considering the experience already gained in operating the plant, the most satisfactory control has been obtained by roughly adjusting a given chamber to the quality of the air obtainable from the central system, and by depending upon the supplementary chamber equipment for the finer degrees of regulation.

For some tests it was found that the main cooler had sufficient capacity to remove not only the sensible heat but also all of the latent heat necessary to be abstracted to reduce the air to the required dew-point. Thus, when the circumstances were favorable the dehumidifier was shut down, and the return air was diverted through the by-pass directly to the central fan.

Automatic Control System

Each chamber is provided with a two-pen wet- and dry-bulb recorder-controller with the bulbs located in the return ducts near the chamber outlets. The control system uses compressed air for actuating the valves and dampers. By means of switches located on separate panels near each chamber the valves and dampers may be permanently opened or closed or connected to the recorder-controllers for automatic operation. Assuming a given chamber to be adjusted for operating with air to be drawn as needed from the central system, and assuming further that the dry-bulb and wet-bulb temperatures within the chamber are at the values set on the controller, then the functioning of the control system may be explained as follows:

(1) With rising dry-bulb temperature and rising wet-bulb temperature:

(a) The wet-bulb controller gradually opens damper 1 to dry air from the central system, and at the same time gradually opens damper 3 to permit air to return to the central system.

(b) The dry-bulb controller gradually opens damper 2 to cool dry air.

(2) With falling dry-bulb temperature and rising wet-bulb temperature:

(a) The wet-bulb controller gradually opens damper 1 to dry air from the central system, and also gradually opens damper 3.

(b) The dry-bulb controller gradually closes damper 2 to cool air and opens it to warm air.

(c) If the dry-bulb temperature continues to fall and damper 2 becomes completely open to warm air, then the dry-bulb controller will turn on the electric heater in the chamber supply duct.

(3) With rising dry-bulb temperature and falling wet-bulb temperature:

(a) The wet-bulb controller gradually closes damper 1 to dry air, and also closes damper 3.

(b) The dry-bulb controller gradually opens damper 2 to cool air.

(c) If the dry-bulb temperature continues to rise and damper 2 becomes wide open to cool air, then the dry-bulb controller will take the control of damper 1 from the wet-bulb controller and gradually open damper 1 to the central

system, and also gradually open damper 3. (This is necessary because the central system is the only source of cool air.)

(d) If the wet-bulb temperature continues to fall, the wet-bulb controller will open the spray valve of the chamber humidifier, and thus offset the dryness of the cool air drawn from the central system.

(4) With falling dry-bulb temperature and falling wet-bulb temperature:

(a) The wet-bulb controller gradually closes dampers 1 and 3.

(b) The dry-bulb controller gradually opens damper 2 to warm air.

(c) If the dry-bulb temperature continues to fall, and damper 2 becomes wide open to warm air, then the dry-bulb controller will turn on the electric heater.

(d) If the wet-bulb temperature continues to fall and damper 1 becomes completely closed to conditioned air, then the wet-bulb controller will turn on the chamber humidifier.

Variations in the Operation of the Chamber Controls

To put a chamber on room air the manual control switches are turned so that damper 5 is opened to the room and closed to the central system; damper 2 is disconnected from the controller and closed to warm air; damper 3 is closed, and damper 4 is put on automatic control so that it may operate in conjunction with damper 1. The room-air method may be used whenever the dry-bulb temperature within the chamber is high enough above room temperature so that cooling can be obtained by natural loss of heat through the walls of the chamber, or by mixing room air with return air from the chamber, provided that the absolute humidity of the room air is less than the humidity maintained within the chamber. The automatic controls operate as already described, except that the room becomes the source of cool-dry air; damper 2 is permanently closed, and heat is supplied by the electric heater. A variation of this method would be the returning of the chamber air to the central system through damper 3 instead of exhausting it through damper 4.

The electric heater of each chamber is divided into three banks of 1000 w (watts) each for the larger chambers and 500 w each for the smaller chambers. Switches are provided so that one or more banks may be put on continuous operation or on automatic control.

For the three larger chambers each humidifier contains three means of adding water vapor; namely, hot or cold water sprays, steam jet, and evaporation of water in a pan. The pan-type humidifier is hand regulated only; and when not being used as a humidifier, the steam-type water heater may be used as a source of heat for the chamber, either alone or to supplement the electric heater. The water sprays and steam jet may operate separately or together; and may be put either on continuous operation or on automatic control. The smaller chambers are equipped with water sprays and steam jet only.

The stops on the damper motors may be adjusted for a maximum or minimum opening to limit the amount of conditioned air drawn into the chamber system. This method is especially useful for those chambers which are operating at intermediate conditions. The stops are set so that the chamber can get a little more air than is needed, but not so close as to interfere with the normal functioning of the automatic controls.

It was pointed out that the dry-bulb controller sometimes operated damper 1. By opening a by-pass valve, damper 1 may be made to operate only through the wet-bulb controller; and for some conditions better regulation has been obtained by this method, especially when it is used with the proper setting of the stops on the damper motors.

Central System Controls

The silica-gel dehumidifier is started and stopped manually. It is in continuous operation whenever it is used. The automatic controls which the unit requires pertain only to the continuous functioning of the unit itself and are not connected with the other controls.

Likewise, the refrigeration unit has its own independent controls. It may run continuously; or it may operate intermittently from a thermostat inserted in the cool air duct leaving the evaporator.

The steam-coil heater and water-type cooler of the central system are regulated by hand.

The respective pressures in the cool and warm air headers are maintained at the same value by means of static pressure regulators which operate the volume control dampers 6 and 7. Damper 8 is connected with damper 7 so that damper 8 can never be opened so wide as to lower the pressure in the cool air header below the value set on the static pressure regulator. However, by adjusting valves in the compressed air lines, damper 8 may be operated independently whenever the circumstances are favorable.

No attempt has been made so far to formulate a set of rules to guide the operators of the plant. With nine chambers in use simultaneously, quite a variety of conditions may be attempted; and the particular method of control adopted for a given chamber, or for the central system, is usually obvious in the specifications of the conditions to be maintained.

Fuels Meeting in Columbus

The third meeting of the Coal Division of the *American Institute of Mining and Metallurgical Engineers* and the Fuels Division of the *American Society of Mechanical Engineers* was held at Columbus, Ohio, October 5-7, in the Deshler-Wallick Hotel with 200 in attendance. R. A. Sherman of Battelle Memorial Institute was chairman of the Joint Program Committee and S. R. Beitler was general chairman of the local Arrangements Committee.

Among the speakers at the technical sessions were R. L. Rowan, Philadelphia, who presented a code for testing coal; the mining of anthracite was described by Prof. John W. Buch, State College, Pa.; the subject of bituminous mining was discussed by Thomas F. Downing, Jr., Philadelphia, and William C. McCulloch, Duquoin, Ill.; the viscosity of coal-ash slags was the subject of a paper by P. Nicholls and W. T. Reid, Bureau of Mines, Pittsburgh.

On Saturday, October 7, an opportunity was afforded for inspection trips of the Jeffrey Manufacturing Co., Columbus Municipal Light, Columbus and Southern Ohio Electric Co. plants, and the laboratories of Battelle Memorial Institute.

The Use of Air-Velocity Meters

By G. L. Tuve*, D. K. Wright, Jr.** (*MEMBERS*) and L. J. Seigel***
Cleveland, Ohio

This paper is the result of research sponsored by the American Society of Heating and Ventilating Engineers in cooperation with Case School of Applied Science

Summary and Conclusions

RESEARCH work on air-velocity instruments conducted at Case School of Applied Science in cooperation with the Committee on Research of the ASHVE, has been in progress for about three years and has proceeded along two main lines: (1) Investigation of methods for instrument calibration; (2) A study of the practical use of instruments for measuring air-stream characteristics and air quantities.

In this report, a comparison is made of several methods of instrument calibration, and calibration curves for typical instruments are shown. Use of the various instruments is also reported, applying to supply openings, air intakes and free air streams. For the free air streams an analysis is based on the law of conservation of energy.

Several instruments are available for air velocity measurement, and calibration curves indicate that the instruments themselves give highly consistent readings. Each instrument has its special advantages, the rotating-vane anemometer being the best averaging instrument, and the bridled-vane anemometer (Velometer) the most convenient to use for *spot readings*. The heated thermometer has the widest velocity range, the heated thermocouple is best adapted for distant reading, and the Kata-thermometer is the least expensive.

Accurate calibrations are quite necessary, and calibration methods for all instruments should be standardized. Instruments may be calibrated in a wind-tunnel air stream having a working section of uniform velocity, located at the discharge of a rounded-entrance nozzle or contraction, but calibration in other types of air streams is not satisfactory because other velocity patterns cannot be duplicated. If a rotating arm is used for calibrations, its length should be at least 10 ft to obtain accuracy within 3 per cent.

For volume measurements at intake or discharge openings, the rotating-vane anemometer is the most convenient, since a traverse may be made and the average velocity obtained in one reading, but a correction factor of 0.70 to 1.05 is usually necessary to compensate for the specific conditions of use. Correction factors for various conditions are given herewith, but a further comprehensive investigation of the use of the anemometer is urgently recommended. It should be possible to develop a set of simple rules and correction factors which will give engineers confidence in this instrument even for measurements where an accuracy within 2 per cent is required.

For measurements of air velocities in a free air stream in a room, the Velometer is the most convenient instrument because of the large number of readings that can be obtained in a short time. The heated-thermometer and hot-wire instruments have possibilities for low velocity work, especially in the laboratory, and further investigations should be made in their application.

Scope of Air-Measurement Problem

Every air conditioning problem involves the moving of the air *to* the conditioned space and the distribution of air *within* the conditioned space. The public evaluates an air conditioning installation almost entirely by the quality of the air distribution, and unless the engineer can make satisfactory air velocity measurements he is risking public dissatisfaction with what may otherwise be an excellent system.

In ventilating and air conditioning work, there are four common occasions for measuring air velocity: (1) In a duct; (2) At the face of a supply outlet; (3) At an intake or suction opening; (4) In an open space or room. Duct velocities are usually high enough to permit of direct impact measurement by the Pitot tube and manometer, while room air velocities range down to the barely perceptible air movement associated with a *draft*. The measuring of an air velocity of 50 fpm is quite different from the measuring of 1000 fpm, and calls for different instrument characteristics. A second problem arises from the fact that the air velocities in various parts of a stream are different, and this *velocity distribution* greatly affects the readings of large instruments such as the common 4-in. anemometer.

In ventilating and air conditioning practice, the measurements of air velocity are usually in the range below 2000 fpm, and the present discussion will be confined to this range. The three purposes of air-velocity measurement in air conditioning work are: (1) For sound-level or noise control; (2) For securing proper air distribution; (3) In order to avoid *drafts*. To satisfy this last requirement, measurements must be made of velocities well below 100 fpm.

In view of these several requirements, the engineer would like further information on the following questions:

1. What are the advantages and what are the limitations of each of the common air-velocity meters?
2. How may air-velocity meters be calibrated, and how accurate are the calibrations?

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***Cleveland Electric Illuminating Co., Cleveland, Ohio.

Presented at the Fall Meeting of the American Society of Heating and Ventilating Engineers, Atlanta, Ga., October, 1939.

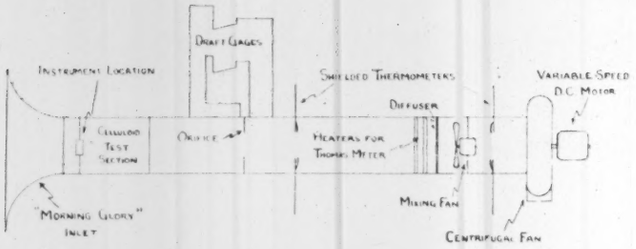
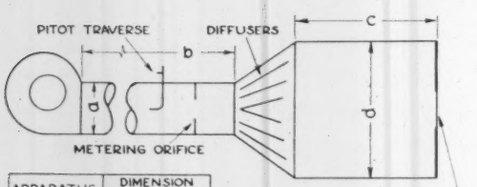


Fig. 1—12-in. calibrating tunnel—apparatus *B*



Apparatus	Dimension			
	a	b	c	d
C	21"	18'	6'	4'
D	18"	11'	4'	4'
E	9"	10'	3'	3'

Fig. 2—Plenum box test units

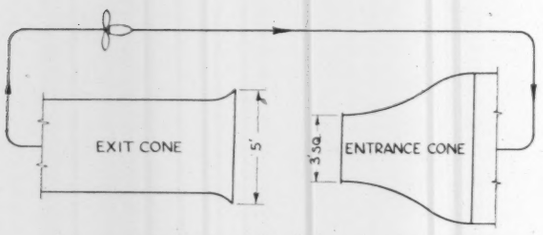


Fig. 3—Test section of Göttingen-type tunnel—apparatus *F*

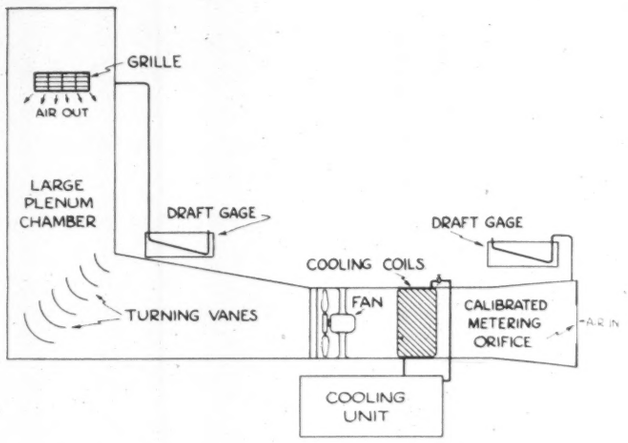


Fig. 4—Test unit for grilles and registers—apparatus *G*

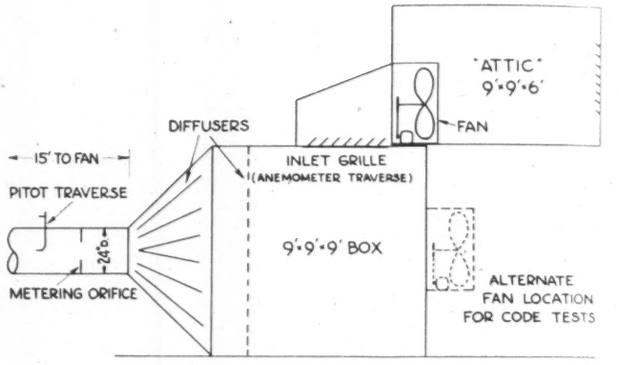


Fig. 5—Test unit for large propeller fans—apparatus *H*

3. What special precautions are necessary for using instruments in air streams: (a) Discharging from supply openings? (b) Entering intake openings? (c) Traversing space in an open room?

The advantages and limitations of the various instruments become apparent when the methods of calibration and methods of application of the instruments are investigated, hence the problem has been attacked from that angle.

Experimental Work

Some of the equipment which was made available for this work is illustrated by the diagrams of Figs. 1 to 5. Fig. 1 is a diagram of a 12-in. diameter wind tunnel of the draw-through type with a large bell-mouth entrance and a transparent working section. This unit was used for instrument calibrations, and the velocity across its working section was strictly uniform as far as could be determined with a Pitot tube, (except within 0.5 in. of the wall.) Three measuring standards were used in this tunnel: (1) A thin-plate orifice; (2) An electric-thermal meter; (3) A Pitot tube. All three of these primary standards agreed very closely, (within 1 per cent at the higher velocities.)

Fig. 2 gives the dimensions of three different metering tunnels of the blow-through type, each with a diffuser and a plenum chamber or still-box. Fig. 3 is a diagram of a Göttingen-type recirculating wind tunnel with a 36-in. square working section, located in the Case Aerodynamics Laboratory. Fig. 4 shows a large plenum-box unit used mainly for the study of supply outlets and grilles. Additional inlet or discharge ducts were attached as required when using the units designated as Apparatus *C*, *D*, *E* and *G* for studies of supply and exhaust grilles. Apparatus *H* of Fig. 5 is a large set-up for comparison of the methods of measuring the air handled by large propeller fans, and by typical attic fan installations.

In addition to the wind-tunnel units, a rotating-arm apparatus (called hereafter Apparatus *A*) was used for anemometer calibrations. Interchangeable rotating arms of 2½ ft, 5 ft and 10 ft lengths were provided, and the unit was equipped with a solenoid starter and electric timer, operated from the same switch.

These eight test units were all used in the experimental work reported herein, and they are identified by letter in the tables and figures which are included. One of the chief methods for insuring accuracy in this work was to repeat the tests on two or more of these pieces of equipment, using different observers. In the use of this test equipment, all measurements of Pitot-tube, orifice, or plenum pressures were made with special inclined manometers which could be read to 0.001 in. water, and which were calibrated to this accuracy over their entire

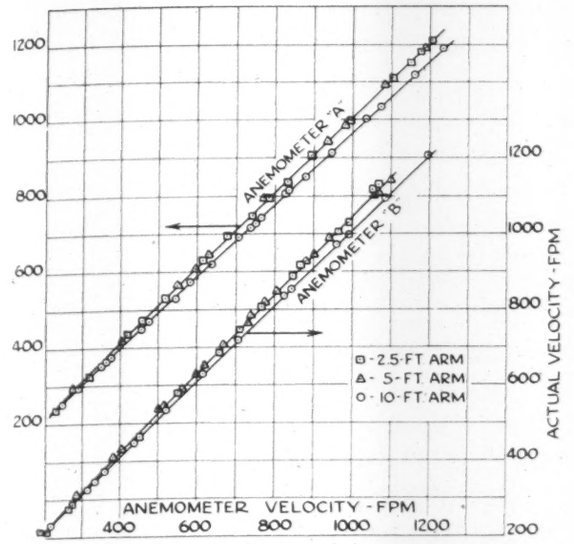


Fig. 6—Anemometer calibration by apparatus *A*, effect of length of rotating arm

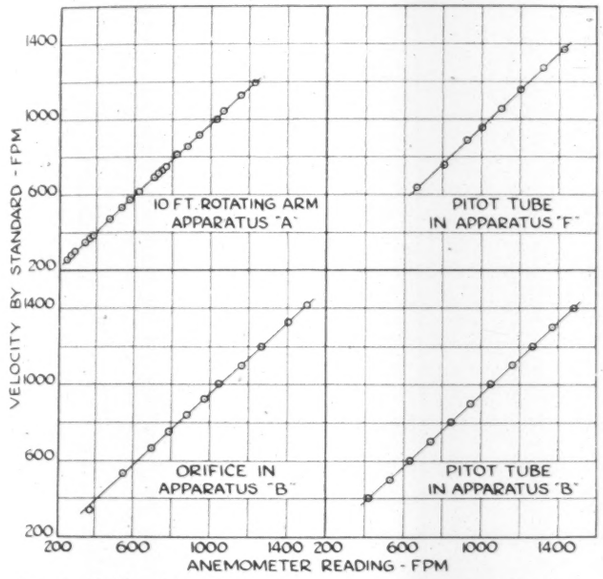

10 FT. ROTATING ARM APPARATUS "A"
PITOT TUBE IN APPARATUS "F"
ORIFICE IN APPARATUS "B"
PITOT TUBE IN APPARATUS "B"
VELOCITY BY STANDARD - FPM
ANEMOMETER READING - FPM

Fig. 7—Calibration of rotating-vane anemometer *A* by four methods

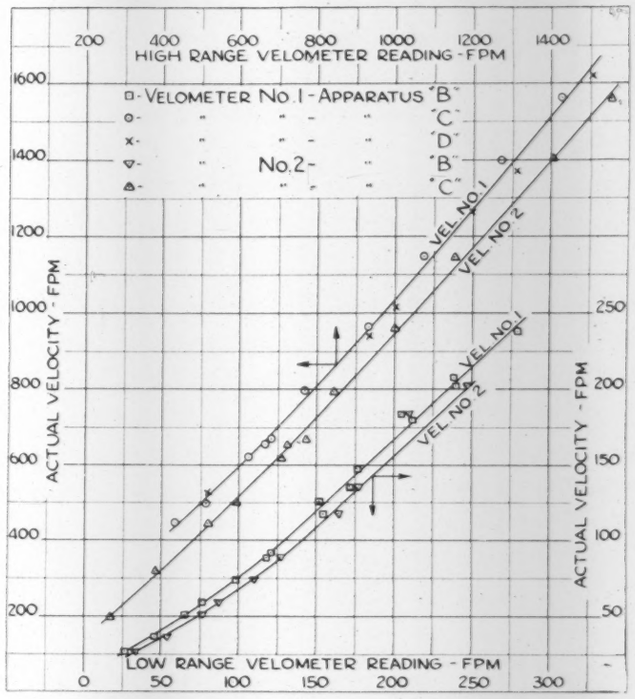

HIGH RANGE VELOMETER READING - FPM
□ - VELOMETER No. 1 - APPARATUS "B"
○ - " " " - " "C"
× - " " " - " "D"
▽ - " No. 2 - " "B"
△ - " " " - " "C"
VEL NO. 1
VEL NO. 2
ACTUAL VELOCITY - FPM
LOW RANGE VELOMETER READING - FPM

Fig. 8—Velometer calibrations

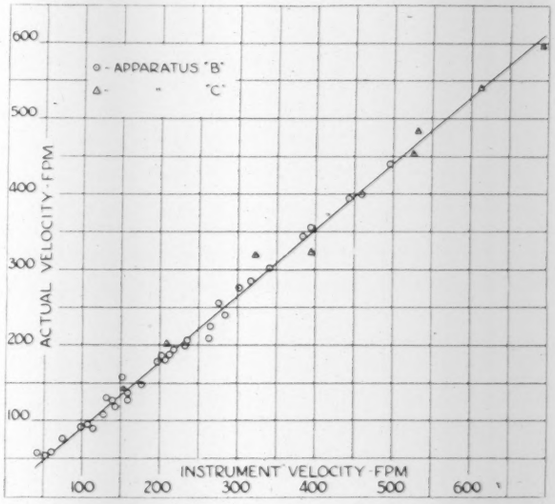


Fig. 9—Heated thermometer anemometer calibration

range by comparison with a sensitive micro-manometer. Special precautions were taken to prevent accidental leakage from ductwork and plenum chambers, and leakage calibrations were made in all cases.

Calibration Methods

Inquiry among the manufacturers indicates that they are using a wide variety of calibration methods. Many instruments are being calibrated by the secondary method of checking one instrument against another one of similar design which has in turn been calibrated by some primary method. One manufacturer indicated that he was using as a standard an instrument which had been calibrated in 1918. In several cases the air stream discharged from a centrifugal fan through an ordinary open-end round duct is being used for the comparison.

There are two classes of calibration methods. One is to move the instrument in still air, the other is to move the air and to fix the instrument in the air stream, comparing its reading with that of another velocity instrument or with a volume measurement by the equation $Q = AV$. The large amount of space and the complicated apparatus required for the straight-line towing of an instrument makes this method generally impractical. Swinging the instrument on a rotating arm is the next best substitute, and for anemometer calibrations, the rotating-arm apparatus is available both at the U. S. Bureau of Standards and at the National Physical Laboratory, London.

An investigation was undertaken in the Case laboratory to ascertain the effect of the length of rotating arm when this method is used for calibrating rotating-vane anemometers. The results for two typical 4-in. anemometers are shown in Fig. 6. Three arm-lengths were used, 30-in., 60-in., and 120-in., and both the instrument control and the timing were accomplished by elec-

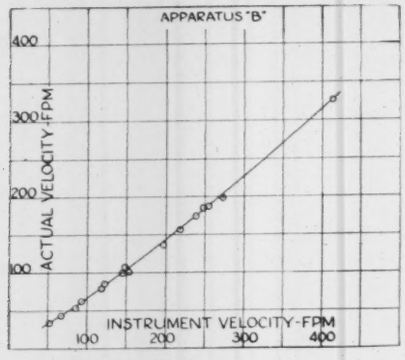


Fig. 10—Heated thermocouple anemometer calibration

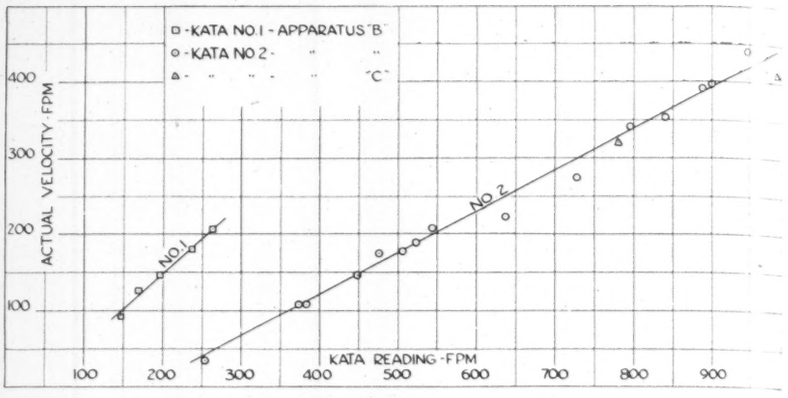


Fig. 11—Kata-thermometer calibrations

trical methods. It will be noted that the two shorter lengths of arm gave almost identical results, but the results with the 10-ft arm agreed much more closely with the wind-tunnel method of calibration, as shown in Fig. 7. If the average of the three wind-tunnel calibrations is taken as the standard, an anemometer reading corrected by the calibration with a 10-ft rotating arm gives a result which is about 3 per cent high, and the shorter rotating arms give a result which is about 7 per cent high. The instrument mounting as well as the rotating arm itself must be of such design as to produce a minimum of air disturbance.

When an instrument is calibrated in a moving air stream, the instrument is more easily read. In fact it would be almost impossible to read some instruments on a rotating arm, (the Kata-thermometer, for instance.) But the moving air stream involves two difficulties. First, a stream of uniform velocity across a large section must be provided, and, second, some primary method of measuring the air-stream velocity must be used as a standard. A third requirement for certain instruments is that the static pressure within the working section must be strictly uniform. Research on aerodynamic wind tunnels has proven that the first and last requirements may be met only when the working section is located at or just beyond the throat of a rounded-entrance nozzle with a large reduction ratio, (see Figs. 1 and 3).

Given a uniform velocity across the working section, several standard methods of velocity measurement are available. In this investigation the orifice, the Pitot tube and the electric-thermal meter were used and the results agreed closely by the three methods. Calibrations of various instruments by the wind-tunnel method, using a nozzle-throat as the working section in all cases, are shown in Figs. 7 to 11.

The consistency of measurements with the rotating-vane anemometer is well illustrated by Fig. 7. It is also concluded that since the same methods were used throughout, the *scattering* of points in Figs. 8, 9 and 11 is due to the instruments and their manipulation, rather than to the calibration methods, since this scattering does not appear in Fig. 7, and practically all observed points were plotted in every case.

The low-range Velometer calibrations are made with the instrument box only, and the high-range calibrations with tube and *jet*. The jet of Velometer No. 1 was a ¼-in. round opening, that of No. 2 was an 0.05-in. x 4-in. slot. A straight-line calibration curve was obtained for the rotating-vane and the heated thermometer anemometers, but the Velometers and the heated thermocouple definitely require curves which are concave upward. The wide-range capacity of the heated thermometer is an outstanding feature, and the same principles could be applied to the heated thermocouple, although the one here calibrated was strictly a low-velocity instrument, (Hukill design.[1])

Kata thermometer No. 1 (Fig. 11) was unfortunately broken after partial calibration, and a duplicate thermometer (No. 2) purchased from the same maker showed entirely different characteristics.

Air Measurements at Discharge Outlets

The character of the air flow at an outlet depends mainly upon the approach conditions immediately upstream, and upon whether the outlet is free-open or covered with a grille or register. There are, of course, infinite varieties of directional-flow outlets, but it is important to understand the straight-flow outlet first, and this discussion will be confined to the latter. There are two extreme types of locations for straight-flow outlets: (1) In the wall of a plenum chamber; (2) At the end of a duct the same size as the outlet.

When air is discharged from a thin-walled plenum chamber or a very large duct, through an ordinary small square-edged outlet, either free-open or covered by a low-resistance grille, there is a jet contraction similar to that obtained with a round, sharp-edged orifice in a thin plate. In addition, the static-to-velocity pressure conversion continues beyond the plane of the opening. These two facts make it very difficult (if not impossible) to obtain a true *average* velocity across the face of such an opening for the accurate application of the continuity equation, $Q = AV$. As a practical solution for this difficulty the anemometer traverse is usually used, with such a *correction factor* as may be necessary. When the air is being discharged from a grille or opening located at the end of a duct of the same nominal size as the grille, the jet contraction at the opening is much less than in the case of a plenum outlet, and hence the anemometer factors are different also.

The 1939 edition of the Society's GUIDE[2] quotes the correction factors obtained by Davies[3] as follows: "for

[1]Hukill, W. V., An Anemometer for Measuring Low Air Velocities (*Refrigerating Engineering*, October, 1934, p. 197).

[2]ASHVE HEATING, VENTILATING, AIR CONDITIONING GUIDE, 1939, p. 525.

[3]Davies, L. E., Measurement of the Flow of Air Through Registers and Grilles. (ASHVE TRANSACTIONS, Vol. 36, 1930, p. 201; Vol. 37, 1931, p. 619; Vol. 39, 1933, p. 373.)

Table 1—Anemometer Traverse Data for Air Outlets and Supply Grilles

Type of Grille	Type of Approach	Velocity Range, fpm	Size, in.	Per Cent Free Area	Number of Tests	Anemometer Factor, F[a] Based on Free Area	Anemometer Factor, F[a] Based on Effective Area
Free-open, no grille	Plenum	300–1600	18 x 6	100	5	0.67	0.67
Free-open, no grille	Plenum	700–1700	12 x 6	100	5	0.73	0.73
Steel-bar grille, H bars on ½-in. centers	Plenum	250–500	12 x 6	78	4	1.07	0.95
Steel-bar grille, V and H bars	Plenum	600–1000	18 x 6	79	4	1.03	0.92
Steel-bar grille, H and V bars	Plenum	250–800	20 x 8	79	9	1.02	0.91
Steel-bar grille, H and V bars, fine mesh	Plenum	400–1400	24 x 8	76	9	1.01	0.88
Free-open, no grille	Small duct	300–900	12 x 6	100	11	1.06	1.06
Steel-bar grille, H bars on ½-in. centers	Small duct	300–1300	12 x 6	78	4	1.14	1.01

[a] F is the anemometer correction factor, defined by the equation: Actual cfm = FAV, *where* A is the area and V is the average velocity by anemometer traverse.

average use, with supply grilles, F can be taken as 0.97 at velocities from 150 to 600 fpm, and as 1.00 at higher velocities." This applies to a uniform traverse, using 6-in. squares, and with the anemometer held against the grille face. For calculating volumes the *effective* area is to be used, and this is defined as the average between the core area and the free area. (Using the definitions given on page 565 of THE GUIDE, 1939). No mention is made of the type of approach. Davies' work was based primarily on small-duct approaches, *i.e.*, an approach duct the same size as the grille.

Table 1 gives the results of over 50 anemometer tests on outlets of various kinds, in which the air was also measured by a Pitot tube or a calibrated orifice. These results show that the anemometer factor for a supply outlet may under certain conditions differ greatly from the values given in THE GUIDE, 1939. These tests indicated that the type of approach had a much greater effect on the coefficient than did the velocity, and that an outlet with a grille in place might behave quite differently from an outlet with no grille. Moreover the use of the *effective area* seems to be a complication of doubtful value.

Traverses of supply outlets may be made with other instruments than the rotating-vane anemometer but they are less convenient. The Velometer with a small *jet* is well suited to measuring the discharge velocity of nozzle-type grilles, because the velocity across the face of each nozzle is practically uniform. For a plenum supply outlet one other method is available, namely the treating of the entire opening as a metering orifice. Table 2 gives the orifice coefficients obtained by applying the common orifice equation to several types of outlets. Such coefficients may be used for calculating the capacity of an outlet when a given supply pressure is available in the plenum, or for obtaining the grille resistance when a given capacity is to be discharged. Manufacturers' resistance tables which deviate greatly from these values should be questioned. The orifice coefficient method is not applicable to openings located at the end of a small duct.

Measurements at Intake Openings

Measurements of air velocity and quantity at the face of an air intake are in some respects more satisfactory than measurements at a discharge outlet. The velocity pattern at an intake is much more uniform, and there are no large variations due to upstream approach conditions. Hence such intake measurements are valuable in connection with tests of fans, unit heaters and coolers, warm-air furnaces and small air conditioners.

For obtaining average velocities which may be used for volume calculations, the rotating-vane anemometer is again the most satisfactory instrument. Davies[4] reported correction factors of 0.762 to 0.832 when using the anemometer against the face of intake grilles of various designs and traversing to obtain an arithmetical average. Greene and Dean[5] derived a correction factor of 0.873 for typical exhaust grilles, when the traverse was made by a special averaging method according to Simpson's rule, giving greater weight to the center readings. The unusual nature of the method proposed by Greene and Dean, in addition to the fact that it is applicable to only a certain range of grille sizes, will limit its use. There is some question whether the anemometer calibration methods were comparable in the two investigations just cited. The 1939 GUIDE used Greene and Dean's coefficient of 0.873 *without* adopting their special traverse method.

It seems that in studies of anemometer measurements at air intakes, two variables of great importance have been given very little consideration. One of these variables is the size and shape of the opening, the other, the edge condition or approach to the opening. Fig. 12 gives the results of an actual test showing the importance of small changes in the edge or approach conditions, even when the openings are large.

The tests reported in Table 3 show that a traverse of a large projecting intake duct does give the low anemometer factors obtained by Davies and others, but if the duct has a flange or if the opening is in a flat wall or ceiling, the coefficient or correction factor is close to unity. As the size of opening is reduced, the factor becomes smaller. The factor varies slightly with velocity as well as with the size, shape and edge condition of the opening.

[4] Loc. Cit. See Note 3.

[5] Greene, A. M., Jr., and Dean, M. H., The Flow of Air Through Exhaust Grilles. (ASHVE TRANSACTIONS, Vol. 44, 1938, p. 387.)

Table 2—Discharge Coefficients for Air Outlets and Supply Grilles Mounted on a Plenum or Large Duct

Type of Grille	Apparatus Used	Velocity Range, fpm	Nominal Size, in.	Core Size, in.	Per Cent Free Area	Number of Tests	Discharge Coefficient, C[a]
Free-open, no grille	*E, G*	400–1400	12 x 6	12 x 6	100	5	0.59
Punched sheet grille, oval openings	*D, E*	200–800	12 x 6	11 x 5	49	8	0.68
Punched sheet grille, square openings	*E, G*	700–1400	12 x 6	11 x 5	51	19	0.73
Assembled strip grille, fine mesh	*E*	400–1400	12 x 6	11 x 5	62	10	0.77
Bar grille, H and V bars on 1 in. centers	*E*	400–1400	12 x 6	11 x 5	66	6	0.77
Bar grille, H and V bars on ½ in. centers	*G*	500–1700	24 x 8	.....	72	10	0.76
Steel sheet with multiple round-entrance nozzles ⅝ in. diameter on 1 in. centers	*D,*	500–2100	.....	12 x 6	30.7	5	0.91
	E, G	500–2200	.....	18 x 6	30.7	5	0.88
		500–2400	.....	24 x 8	30.7	8	0.91
		500–2300	.....	12 x 12	30.7	7	0.89

[a] C is defined by the equation: cfm = $1096AC\sqrt{h/d}$, *where* h is the upstream total pressure in inches of water, d is the air density, and A is the free area or *daylight* area.

Note: Logarithmic graphs of these and other grille tests show an almost exact conformity to the square-root law.

Table 3—Anemometer Traverse Data on Rectangular Intake Openings
(Free openings, no grilles)

Type of Intake	Size, in.	Face Velocity, fpm	Apparatus Used	Number of Tests	Anemometer Factor, F_a
End of duct (no flange)	14 x 5	500–1700	*G*	5	0.73
End of duct (no flange)	14 x 8	500–1500	*G*	5	0.73
End of duct (no flange)	14 x 10	500–1800	*G*	6	0.77
End of duct (no flange)	24 x 24	400–500	*G*	2	0.80
End of duct (no flange)	30 x 42	615	*H*	2	0.89
End of duct (no flange)	36 x 50	580	*H*	6	0.89
Square-edged wall opening	14 x 8	500–1500	*G*	7	0.83
Square-edged wall opening	24 x 24	400	*G*	3	0.92
Square-edged wall opening	30 x 42	515	*H*	2	1.03
Square-edged wall opening	36 x 50	575	*H*	4	1.03
Round-entrance wall opening	30 x 42	530	*H*	2	1.01
Propeller fan casings (square duct)	32 x 32 to 45 x 45	500–1200	.....	18	Avg 0.90

[a]*F* is the anemometer correction factor, defined by the equation: Actual cfm = *FAV*, *where A* is the total area traversed and *V* is the average velocity by anemometer reading.

Still another variable is the method of traverse. All data given in Table 3 and Fig. 12 were taken with a 4-in. anemometer, (though both 3-in. and 6-in. anemometers were used for check). Areas were divided into rectangles 4 in. to 5 in. on each side, and all traverses were simple cumulative readings, using 5 seconds to 30 seconds in each rectangle, depending on over-all size.

It is evident that the GUIDE recommendations for the use of anemometers give only approximate results as they stand at present, and that additional studies should be made to furnish accurate factors for all common conditions.

Air-Stream Measurements in Open Rooms

Proper solution of the problem of air distribution in a room requires a knowledge of the spread and the throw of the entering air stream, and of the velocity distribution within it.

The included angle of spread of a straight-flow stream usually falls within the range of 12 deg to 25 deg for jets of air, steam and other fluids under a wide variety of conditions. Of course almost any desired angle larger than this may be produced by directional guide vanes, although the actual resulting stream angle is less than the angle of the vanes, and for uniform distribution the vane angles must be graduated across the face of the opening. Van Alsburg,[6] Mackey,[7] Jordan,[8] Theiss,[9] and others have reported values of the angle of spread from a straight-flow grille, and from such values a scale layout may readily be made for any room.

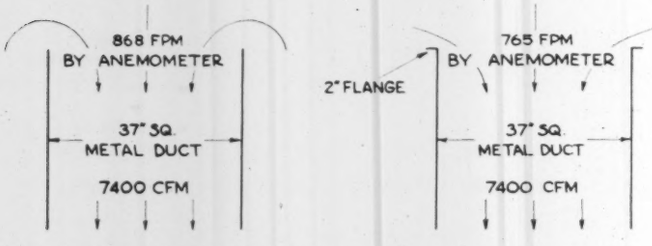


Fig. 12—Effect of edge conditions on intake anemometer traverse

[6]Van Alsburg, J. H., Characteristics of Registers and Grilles. (ASHVE TRANSACTIONS, Vol. 41, 1935, p. 245.)

[7]Mackey, C. O., The Rationale of Air Distribution and Grille Performance. (*Refrigerating Engineering*, June 1938, p. 417.)

[8]Jordan, R. C., Air Diffusion in Ventilation Systems. (ASHVE JOURNAL SECTION, *Heating, Piping and Air Conditioning*, May 1935, p. 253.)

[9]Theiss, An Experimental Study of Air Entrainment in Room Ventilation, master's thesis, Case School of Applied Science.

The throw of the stream depends on the angle of spread and on the initial velocity and quantity of air. The ceiling or the floor will interfere with the throw of warm or cold air streams. Maximum throw of the stream is usually defined as the normal distance from the face of the outlet to the plane in which the maximum velocity does not exceed 50 fpm. Experimental studies have shown that:

1. Maximum throw is directly proportional to average velocity at the supply outlet or grille.
2. Maximum throw is proportional to the square root of the free area of the supply outlet or grille.

These laws have been stated by Greenlaw and Hart[10] and others, and tests in the laboratories of Case School of Applied Science[11] indicate that they may be applied to most types of air conditioning grilles. Several other laws may be derived from those already given, and each of these laws aids in checking the accuracy of instrument applications and test methods.

Another important principle applying to free air streams is that of conservation of momentum, which

Table 4—Results of Air-Stream Analysis by the Conservation Law

Type of Outlet	Free Opening in Plenum	Open End Duct
Size of outlet, in.	14 x 8	14 x 8
Rate of flow, cfm	1161	1156
Horizontal spread, total angle, degrees	23.5	24.5
Measurements in a normal plane 3 ft from face (see Fig. 13):		
Mean velocity, fpm	367	316
Cross-sectional area of stream, sq ft	7.3	7.8
Air entrainment, per cent of primary air	478	347
Summation of momenta in 6 in. squares	176,600	117,600
Measurements in a normal plane 6 ft from face:		
Mean velocity, fpm	327	250
Area of stream, sq ft	13.3	14.0
Entrainment, per cent of primary air	567	472
Summation of momenta in 6 in. squares	181,700	118,900
Measurements in a normal plane 12 ft from face:		
Mean velocity, fpm	238	202
Area of stream, sq ft	28.2	27.0
Entrainment, per cent of primary air	800	588
Summation of momenta in 6 in. squares	178,700	115,400

states that the product of the mass of the air moving times its average velocity across a normal plane, must be constant. This principle has been frequently used in such jet investigations as those of gas burners,[12] locomotive front ends,[13] and jet pumps. The principle is the same as the law of conservation of energy. From the conservation law both the throw of the stream, and the amount of room-air entrainment by it, may be calculated. Mackey[14] has calculated that the air entrainment is 11 to 1 at the end-plane of the throw. This is based on assumptions of uniform velocity across the stream, 14 deg spread, and 600 fpm initial velocity across an area of one square foot. But the assumption of uniform velocity is in error, and a more accurate experimental verification of the conservation law is possible by using an indicating-ribbon grid and a Velometer, as shown in Fig. 13. Table 4 gives results of such an analysis by Theiss,[15] for two of the seven straight-flow outlets which he investigated. These results, involving thousands of observations and calculations, demonstrate

[10]Greenlaw, A. L., and Hart, T. S., Air Paths from Grilles. (*Heating, Piping and Air Conditioning*, June 1938, p. 387; July 1938, p. 452.)

[11]Tuve, G. L., Measuring Air Distribution and Grille Performance in Air Conditioning. (ASHVE JOURNAL SECTION, *Heating, Piping and Air Conditioning*, November 1937, p. 700.)

[12]Bureau of Standards Technologic Paper No. 193, Design of Atmospheric Gas Burners.

[13]Young, A Study of the Locomotive Front End. (University of Illinois, Engineering Experiment Station Bulletin No. 256.)

[14]Loc. Cit. See Note 7.

[15]Loc. Cit. See Note 9.

Fig. 13—Method of traversing free air streams

the reliability of the method, and of the Velometer, for air-stream measurements in rooms. The difference in air entrainment in the two cases of Table 4 is noteworthy.

In considering the use of other instruments than the bridled-vane anemometer (Velometer) for measuring air velocities in open rooms, the following suggestions may be helpful:

1. Where directional effects are important, use a mechanical-type instrument.
2. Where comfort effects are important, and ample time is available, use a heated-type instrument.
3. For relative measurements, adhere to a single instrument, selecting one that will cover the entire velocity range.
4. For absolute measurements (*i.e.*, those in which an accurate result in feet per minute is required), check the measurements by using more than one type of instrument.
5. Always take the average of several readings to represent the final result.

Construction Industry Conference

The Chamber of Commerce of the United States announces that a Construction Industry Conference will be held at the National Chamber's Building, Washington, D. C., November 16 and 17. At this two day conference the subjects of discussion will be Construction and American Progress, the Market for Homes in the Average Community, Residential and Non-Residential Construction, Jurisdictional Disputes, and the Effects of War on Construction.

The chairmen of the sessions will be E. P. Palmer, New York, George W. West, Atlanta, Ga., Ernest T. Trigg, Washington, D.C., Albert B. Tibbets, New York, and William Stanley Parker, Boston, Mass. Among the speakers will be W. A. Klinger, Sioux City, Ia., on New Horizons; and I. N. Tate, St. Paul, Minn., on Better Living Standards.

Preview of Fall Meeting Papers*

Food Preservation in the South

By C. T. Baker

Food preservation, which involves the control of temperature and humidity with special emphasis on the preservation of food commodities in the South, is described. An outline of the methods employed is given, as well as descriptions of typical refrigerating equipment and of plants for quick-freezing of vegetables, fish and the storage of pecans, one of Georgia's most valuable crops.

A Study of Certain Factors That Affect the Droop Characteristics of Thermostats

By B. E. Shaw and LaVerne Lyon

The paper deals with a subject about which little is commonly known and about which very little has been published. It describes the factors affecting the droop characteristics of thermostats and also describes test apparatus used for studying the influence of cold walls. It is concluded that causes of droop are primarily the slow infiltration of air and thermal inertia of metallic elements. The use of artificial heat in the thermostat and the effect of cold walls on the droop are opposite to that of the factors of thermostatic lag and artificial heat.

Agriculture Has Engineering Problems in Heating and Ventilating

By F. B. Lanham

This paper reviews the agricultural problems which involve heating, ventilating and air conditioning practices and gives some indication of the important contributions that can be made by engineers to comfort and efficiency on the farm. It includes a discussion of the requirements of rural housing, the storing and curing of sweet potatoes and other vegetables, dairy production, hay curing, and stresses the importance of using modern engineering practices for the preservation of farm products.

Permissible Relative Humidities in Humidified Buildings

By Paul D. Close

Two important problems which accompany humidification of buildings for winter comfort are discussed in this paper, and a review of the fundamentals is included. Calculations necessary to provide proper resistance in wall or roof structures to maintain inside surface temperatures to prevent surface condensation are given. A simple chart for obtaining relatively easy approximation of permissible relative humidities at various temperatures is presented. It is applicable for roof or wall constructions for a wide range of conditions.

*To be published in December ASHVE Journal Section.

The Application of Storage Refrigeration to Air Conditioning

By Carl F. Boester* (*MEMBER*), St. Louis, Mo.

THE idea of accumulating or storing up refrigeration over a given period of time for later use is not new. As a matter of fact, the idea has been experimented with for so long that it is impossible to ascertain who should rightfully be given credit for it. In recent years, however, considerable research on design and development has been done in this line with a view to applying the principle to air conditioning, primarily for the purpose of lowering first costs, and secondly, for the purpose of lowering operating costs of air conditioning.

Several methods of applying storage refrigeration to air conditioning have been developed and reported by Leopold, Friend, Freund, Betz, Elmer, Marston and Pfeiffer. The research activities of these men have been along various lines, such as the storage of refrigeration in the form of chilled water, chilled brine and ice. Reports have been made in technical and trade publications. The author's experience has been along the line of accumulating refrigeration in the form of ice on evaporator coils.

Whether or not storage refrigeration is applicable to a particular air conditioning problem depends upon the load. To begin with, the load is determined in the usual manner for any type of installation, using the engineer's own method of heat gain calculation. After the size of the job in tons of refrigeration is determined, the next step is to determine whether or not storage refrigeration is desirable from an application engineering standpoint and, second, whether or not it is economical.

The average commercial or industrial air conditioning load operates only a few hours out of each 24-hour period, and during this time refrigeration is required at varying rates from zero to peak capacity. This brings in two factors—hours of operation and usage or load factor, giving an equation for determining the amount of refrigeration to be consumed, which may be expressed as follows: tonnage (as indicated by the heat gain survey), times hours of operation, times usage or load factor, divided by 24—the result being the amount of refrigeration to be consumed as distinguished from the rate at which it is to be delivered.

Since one ton of refrigeration may be assumed as the equivalent of the production of one ton of ice in 24 hours in a storage refrigeration application of this type, the result of the equation is the equivalent of the total tons of ice that will be consumed and that will, therefore, have to be provided. The refrigerating equipment applying this storage principle must, then, be selected to store up this amount of refrigeration in the form of ice.

Before discussing the means of accumulating the required refrigeration in the form of ice, it is well to consider the factors affecting the equation. To begin with, the tonnage as shown by the heat gain survey expresses the maximum rate at which refrigeration is required for the air conditioning load, and any storage refrigeration system has to be able to deliver refrigeration at the maximum rate as well as at any percentage of this rate, according to particular load requirements. The hours of operation affect the amount of refrigeration to be consumed. The ratio of operating period to accumulating period determines the feasibility of storage refrigeration. Thus, if the accumulating period is 24 hours and the place of business to be cooled requires cooling for eight hours each day, then there is a ratio of three to one. The guiding principle might be expressed as follows: the longer the hours during which cooling is required as compared to the accumulating or recovery period, the less desirable storage refrigeration becomes. If the period over which refrigeration can be accumulated is 72, or possibly 144 hours (as in storing refrigeration for church cooling), the applicability of storage is most apparent; also if the accumulating period is limited to the hours from 6 p. m. to 6 a. m. because of an attractive off-peak rate offered by the power company during this period, the principle may still be applicable.

The other governing factor in the applicability of storage refrigeration is that of usage, generally expressed as *percentage of peak*. For example, in theater air condi-

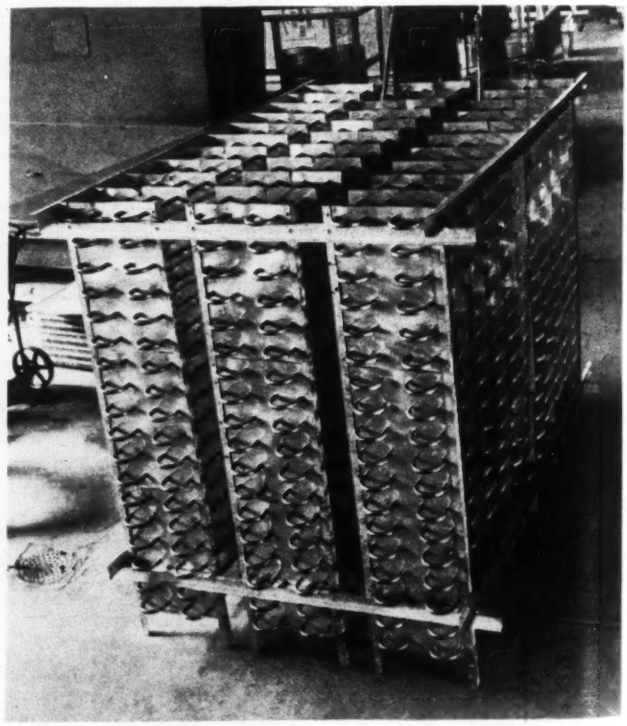


Fig. 1—Typical 3-ton ice accumulating evaporator for storage refrigeration purposes

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Presented at the Fall Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Atlanta, Ga., October, 1939.

tioning, since people comprise most of the load and since the number of people in the theater varies, the usage or load factor also varies as much as from 20 to 90 per cent of the peak or design load. Careful analysis of theater attendance records indicates that for a theater operating 12 hours a day, the usage factor is approximately 55 per cent. For theaters operating only during the evening, the usage factor is 65 to 70 per cent.

Thus, if the hours of operation are long but the usage factor is low (as is often the case in cafeterias and restaurants), or if the hours of operation are short but the usage factor is high (as in the case of church cooling, or, in the case of neighborhood theater operation), then storage refrigeration is quite desirable. When the usage factor is high and the hours of operation are long, it is not practical to design an air conditioning system employing storage refrigeration.

It might be well to mention that determining the usage factor sometimes presents difficulties, but, in addition to the expedient of making a personal survey, it can often be obtained from manufacturers of equipment for storage refrigeration applications.

Assuming, now, that the equation indicates that a given set of conditions makes it desirable to employ storage refrigeration, the engineer is presented with the problem of determining the best way to select and design equipment for the application. If, after analysis, it is considered desirable to store the refrigeration in the form of ice rather than in the form of brine or chilled water, a simple way to accumulate ice is upon evaporator coils submerged in a tank of water.

In considering the tank for this purpose, the question of insulation immediately arises, and whether or not insulation should be used depends upon an analysis of tank losses as compared with insulation costs. This must be determined by the application engineer in arriving at his total cost. Often the tank is placed in the ground where the temperature the year round rarely exceeds 55 F. Thus the temperature difference between the ground and the ice is rather small, and this, multiplied by the square feet of tank surface will give the tank loss.

It has long been known that approximately 80 to 100 sq ft of evaporator surface times the coefficient of heat transfer is required to accumulate one ton of ice at an economical cost. It can be appreciated that the thicker the ice on the evaporator, the lower the refrigerant temperature in the evaporator must be in order to accomplish the desired rate of accumulation and counteract the effect of the accumulated ice acting as insulation. Research indicates that an accumulation to a thickness greater than 1½ in. to 1¾ in. on any type of evaporator surface is not economical.

In designing an evaporator with the required amount of surface, the following must be borne in mind: the amount of refrigerant charge, the refrigerant temperature, the utilization of all of the evaporator surface and the progressive use of such surface, the design of the evaporator to withstand expansion of the ice as it freezes, the weight of the ice on the evaporator section, and the displacement of the evaporator itself and of the evaporator with the required amount of ice accumulated on it in ratio to tank volume.

From the standpoint of lower first cost and lower operating cost, it has been found practical to design an evaporator approximately as outlined. It should be constructed of ⅝-in. tubing because the necessary prime surface in tubing of this size will displace the minimum amount of water in a tank, and thereby require a tank of smaller capacity. For example, 80 sq ft of surface will require approximately 500 lineal feet of ⅝-in. tubing. This might well be made up into an evaporator with tubing on 3½-in. centers having over-all dimensions of 14 in. wide, 63 in. high and 84 in. long. An evaporator of the same capacity made in 1¼-in. tubing would be some 20 in. wide, some 10 ft long and some 80 in. high, and would require a tank having twice the volume of the evaporator using smaller tubing. Such an evaporator almost doubles the tank cost.

The smaller size tubing has another advantage. A ton of ice has a displacement of about 37 cu ft. The more nearly an evaporator coil approaches this volume in accumulating a ton of ice, the more desirable it is, for as the ice is reduced to low water temperatures, the volume of water in the tank becomes disproportionate to the volume of ice in the tank. When this occurs, it becomes extremely difficult to maintain and control low water temperatures.

The method of feeding the refrigerant is of great importance from the standpoint of minimum refrigerant charge. The greatest efficiency has been obtained by the use of multiport expansion valves set at three or four degree superheat. This low superheat setting does not permit the use of any of the evaporator section as a drier; therefore, a heat exchanger is required to accomplish this function. The refrigerant charge and method of feed should be such that the ice is accumulated progressively on the evaporator section, and under such progressive accumulation, it is possible to superheat the refrigerant vapor in portions of the evaporator on which there is no ice accumulation, thereby maintaining higher and more efficient suction temperatures conducive to low operating cost.

An evaporator made of light ⅝-in. tubing is able to flex with the expansion of the ice as it freezes without damage to the evaporator, and the evaporator with its accumulation of ice upon it, has a tendency to float in the tank of water due to the fact that the ice is lighter than water. Therefore, the evaporator itself does not have to bear the weight of the ice and is sufficiently strong even though of such light material.

The most important factor governing the successful application of storage refrigeration to air conditioning is the ability to melt the ice and to control its melting rate according to the load demand. It is at this point that experience in this connection is valuable. While anyone can accumulate ice with varying degrees of success, the ability to melt it at the required rate is all-important. It is quite possible to have a tank two-thirds full of ice and yet withdraw water from the tank at a temperature of 60 to 65 F, entirely useless for air conditioning purposes. The volume of water circulated through the tank, the temperature range over which the water is to be cooled, the velocity of the water through the tank, the control of the water flow through the tank and the number of square feet of ice surface exposed to the flow of the water, very definitely affect the ability to control the rate of melting, and thereby the success of the storage refrigeration application. Data pertaining to

the rate of heat transfer (K) between the ice and the water under various water temperatures, water velocities and square feet of ice surface required to accomplish this result have, as a rule, been considered a trade secret of the manufacturers supplying this type of equipment. The real success of a storage system hinges on these factors. Too rapid melting and thereby too low a temperature wastes refrigeration, and too high a temperature, of course, will not produce the desired results. The manufacturer of ice accumulating evaporators usually breaks the evaporator up into a number of sections in order to provide the correct amount of surface for a given flow of water, and should be consulted on problems of this nature and relied upon for guarantees of performance under given load conditions.

As to operating costs, a ton of ice can be accumulated with approximately 22 kwhr of electrical energy. This is comparable to a ton of refrigeration delivered by other means. The storage refrigeration equipment functioning at high pressure operates at a constant load and as a rule takes advantage of lower condensing temperatures available at night, if a cooling tower or an evaporative condenser is used. On the other hand, a conventional direct-expansion job with frequent cycling and partial load inefficiencies uses the same amount of power. Thus, where no demand charge is made, the power costs of direct expansion and of storage refrigeration are comparable. However, where an off-peak rate is available, or a high demand charge exists, storage refrigeration reflects a considerable saving in operating cost.

As to first cost for the equipment, savings when employing the storage principle have been as high as 70 per cent, and they average about 25 to 30 per cent. On those jobs where hours of operation are long and the usage factor high, storage refrigeration is hardly feasible, and for such a job the first cost would be high.

Storage refrigeration permits many new application engineering practices which are conducive to low first costs. These are too numerous to mention but have become obvious when a comparison of costs is made for a given job. Each individual job must be analyzed to establish whether or not it can best be handled by the application of storage refrigeration or done with full capacity, direct expansion equipment.

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Committee Appointments

Announcement has been made by Pres. J. F. McIntire of the appointment of N. A. Hollister, Brooklyn, N. Y., as ASHVE representative on the Air Conditioning and Blower Systems Committee of the *National Fire Protection Association* to succeed the late Claude A. Bulkeley. A meeting of the Air Conditioning Committee was held at 85 John St., New York, N. Y., October 20, to plan activities for the coming year.

Chairman W. L. Fleisher of the Committee on Research has made the following appointments: R. A. Sherman as chairman of the Technical Advisory Committee on Solid Fuels; R. T. Miller as chairman of the Technical Advisory Committee on Insulation, and as members of the Technical Advisory Committee on Sensations of Comfort, L. C. Soule, Syracuse, N. Y., and C. R. Bellamy, New York City, who is chairman of the Air Conditioning Committee of the *American Gas Association*.

Spray Painting Booth Designed for Safety

A pneumatic spray painting exhaust booth designed to insure health and safety is described by Theodore Hatch and R. E. Northrup, in the Industrial Bulletin of the State of New York. Mist resulting from the use of the paint spray nozzle does not frequently reach the surface but is scattered in the atmosphere. In addition some of the droplets rebound from the painted surface with the conveying air and thus add to the atmospheric contamination. The importance of this contamination from the standpoint of health and safety depends upon the composition of the spray. It may include lead-containing pigments or other harmful ingredients and in the case of lacquers, the solvent may contain benzol or other toxic organic materials. In the absence of highly poisonous compounds from the spray it still contains substances that must not be breathed in the quantities that are released by spray painting. Furthermore, the solvents are usually inflammable and explosive, thus introducing a safety hazard. Therefore, it is necessary to confine the spray painting operation within an enclosure or booth which is constantly ventilated for the removal of the mist and vapors produced and in addition certain fundamental precautions must be taken against the fire and explosion hazard.

The location of the operator as well as the size and shape of objects to be sprayed, are important factors in determining the size of the booth. If the operator stands outside, as in the case of a bench booth, the cross-sectional dimensions are made as small as possible, consistent with the work and the only dimension of importance is the depth of the booth. The rate of ventilation to be maintained in a spray booth is determined by the air velocity through the zone of operation required to prevent the escape of the contaminating substance into the general room air. For most work, the minimum permissible air velocity is 100 lfm. It should be increased to not less than 150 lfm when lead, benzol silica or other toxic substances are present in the spray. When the operator works inside the booth, the minimum air velocity must be created at his breathing zone and the direction of air flow should be horizontally through the booth. The air flow through the booth needs to be uniformly distributed so as to insure the same air velocity in all sections of the operating area.

The rate of ventilation through spray booths is relatively large, and on continuous production operations it is often desirable to deliver fresh air direct to the enclosed spray booth, providing air filters, heaters, fan and piping for this purpose as part of the complete installation.

An Analysis of the Air Conditioning Contractor's Business Relationships

By John H. Carter* (*MEMBER*), St. Louis, Mo.

THE results of a study of business management problems associated with the contracting phases of air conditioning work have been carefully compiled in the belief that a discussion of the subject will be helpful to members. The scope of this study will be limited to the contractor's external relationships with customers and suppliers and will leave the problems of internal organization for later consideration. In general the typical organization under consideration is that of a contractor-distributor who handles primarily the engineered type of job, although some attention will be devoted to merchandising problems. Franchised relationships with suppliers will be analyzed, and compared with the free-lance method of operation.

The typical organization has a personnel of from two to five men, with one or two clerical employees, and extra draftsmen, installation workmen, etc. The principal members are considered as specialists in their own fields, but have certain overlapping duties.

This paper cannot presume to recommend any standard way of organizing or conducting such a business as each organization will have its own problems, which must be solved individually. Many of these problems may not have been foreseen or understood, so an analysis of some of the problems will be made.

Engineering may be considered the link between science and economics. The air conditioning contractor, representing the field of engineering, must apply the findings of science to the needs of the public and translate the language of theory into the language of costs and earning power. He must carefully keep in a middle ground. If he concentrates on business to the exclusion of science, he will have costly technical troubles. If he is too much concerned with technical matters, and neglects proper business management, he may likewise lose money.

The period since the beginning of 1937 has clearly revealed some serious weaknesses in the air conditioning industry. The volume of business in the year 1937 exceeded all previous records, yet some technically competent contractors suffered financial losses during that period even though they had a record sales volume. The year 1938 showed a sharp drop in the volume of air conditioning business, and produced a severe turnover of contractor-distributor franchises, as well as the discontinuance of many such organizations.

Relations with Customers

A common difficulty in customer relations often occurs where no consulting engineer is retained because there has been no "meeting of minds," lacking which a contract legally does not exist. If the contractor or his sales representative is afraid to enter into a comprehensive agreement with the customer at the time the sale is negotiated, the way may be left open for misunderstandings and delays in acceptance of the installation, which are expensive as final payment is held up until adjustments are completed to the customer's satisfaction, possibly at a cost much in excess of that contemplated by the contractor when he priced the job. Such difficulties often arise from loosely-drawn contract phrases and from commitments incorrectly implied beyond those of the written specifications.

When a consulting engineer is not employed, there will be plenty of argument between the customer and contractor, and the best time to have it is at the time of negotiating the contract rather than during the installation. It is advisable to draw up a complete, specific and written agreement covering the inside and outside design conditions, the human, electric, ventilation and other loads, etc. It would be well, too, to mention that "the equipment shall be *capable* of maintaining" the specified conditions "when operated in accordance with instructions furnished by the contractor and the manufacturers." The customer's operating man may be of an experimental turn of mind, but let him play around after the job is paid for, not before.

The methods of acceptance and of payment should be clearly defined. These are fundamentals, and should not be left in question. Unless the contractor has an excess of working capital, he would better write his contracts to have 85 to 90 per cent of the price paid by the time the equipment is installed, connected and running, with the balance 30 days after this date. Minor adjustments of scoops, grilles and control system can go on for a long time, and it is unfair to the contractor to make any considerable final payment wait upon these adjustments, especially since most control systems carry a year's adjustment and service by the manufacturer.

The written contract should be specific as to the extent of the contractor's responsibility for adjustments, repairs, exchanges of defective equipment, replacement of lost refrigerant, and other matters of service. These things are a gamble, but they must be paid for by someone when they are needed. If the contractor knowingly takes this risk, he will allow for it by a certain percentage in the price. If he simply passes along the manufacturers' warranties, covering exchange of defective parts at the factories, he should make sure that the customer agrees to pay for any labor, transportation charges, etc., which will be involved.

There should be stipulations as to the exact degree of completion required for acceptance, and as to requirements for cooperation between various contractors where alterations or other work are going on at the same time.

*Special representative, Frick Co.

Presented at the Fall Meeting of the American Society of Heating and Ventilating Engineers, Atlanta, Ga., October, 1939.

especially where there is a penalty-bonus clause based on date of completion. Adequate elevator service for freight should be arranged, as should lights, telephone service, access to driveways, tool and material storage, field office space, etc. It should be understood particularly what parts of the work, if any, must be done outside of normal working hours on an overtime basis. Some department store jobs call for precautions against pilferage, and of course against dirtying or damaging merchandise, floor coverings, etc. Where scaffolding must be taken down after each night's work, it can involve considerable cost.

It should be kept in mind that a penalty clause for failure to complete at a stated time has no legal force if it is not offset by a specified bonus for early completion. The penalty and bonus rates per day need not be the same, but the bonus rate must be more than nominal. In the same way, on a cost-plus contract with a guaranteed maximum cost, the contractor must be able to participate in savings below the guarantee, if he is to be held to contribute toward any excess of cost over the guarantee.

Some types of customer do not particularly need the services of a consulting engineer, where the customer's staff includes engineers who may have devoted considerable study to the subject. The author knows of some large concerns which have bought air conditioning on this basis, but except in rare cases the job would have been a better one, with more satisfaction and economy to both customer and contractor, if a good consulting engineer had been called in. Certainly the average customer, such as the owner of a hotel, department store, office building, etc., is in no position to compare and evaluate independent bids from various contractors, covering equipment of widely varying design and suitability. It often works out that the several contractors prepare their individual layouts and proposals as they see fit, and then are asked to alter their layouts to conform to a common average which will include the best features of all the systems proposed,—in the opinion of the customer. He may be wrong in this judgment. Whether he is or not, this method represents a needless waste to the bidders.

The Consultant's Role

The advantages to both customer and contractor of having a fair and capable consulting engineer in charge of design, installation, testing and acceptance cannot be over-emphasized. This statement may seem superfluous, but there are still too many architects, owners and contractors who cheerfully invite the responsibility of an unsupervised job, only to find endless delays, arguments, adjustments and changes which hold off the final payment and increase the cost.

The main advantage to the air conditioning contractor is that he has a limited and clearly defined responsibility, and that he can complete the job quickly, collect his money, and get out. This assumes of course that both contractor and engineer are competent. Furthermore, the cost of bidding and selling the job is held down, where design, specifications and method of installation have been drawn up in a clear and definite way.

Some consulting engineers have a reputation for severity and extremely strict adherence to the letter of the contract. The author cannot extend much sympathy to the contractor who loses money because of this characteristic. The various consulting engineers and their habits are pretty well known throughout the industry. Specifications are meant to be read and followed. It is quite proper that the irresponsible contractor learn his lesson the hard way, if he takes chances, cuts corners, estimates carelessly, furnishes substandard equipment, or cheapens the work improperly. The next time he will put in his bid with proper allowances, and will sell in fair competition with his fellow-bidders.

If the plans and specifications, especially the conditions governing acceptance and payment, seem improper or prejudicial, the contractor can enter protest with the customer and the engineer, and as a last resort can always refuse to bid. This may seem like a feeble method of protest, but it really carries weight. No consulting engineer can afford to have his jobs turned down by contractors of good standing, representing good lines of equipment.

Meticulous wording and strict enforcement work both ways. If the contractor meets the letter of the specifications, he is entitled to prompt payment for his work, even though the consulting engineer may have made some mistakes or overlooked some necessary details. The cost of changes requiring extra work or extra material must be paid by the customer to the contractor, on a previously arranged basis of percentage mark-up over cost. It should be specified that the engineer is to inspect all work, where required, promptly upon notification from the contractor.

Government Contracts

Sales to governmental agencies are in a class by themselves. A contractor may be willing to go beyond his legal obligations to install a better job for his regular customer, or to help out a consulting engineer who needs some extras, in the belief that such favors will help him on later jobs. He cannot expect to earn future favors in this friendly way, when he is handling a government contract. In general each job is judged strictly by the exact letter of the contract, and work which betters the requirements cannot help in future competitive bidding,—in some cases, in fact, it may be rejected. The engineering is usually competent, and the specifications usually complete and inflexible.

The contractor should figure on a large amount of paper work. In some cases he must allow for delays in acceptance and payment. These are often caused by departmental red tape, sometimes by a rigid insistence on slight details of the specifications. The inspector is probably well able to see that a certain change makes a better job, and is convinced in his own mind that it should be accepted and paid for, but he usually cannot or will not use his independent judgment, especially if he is the type of government employee who has lost his initiative in the security of routine.

Contractor's Relations with His Suppliers

For the purposes of this analysis, the contractor-distributor is assumed to have, or to intend to get, a resale

contract or franchise with a manufacturer of some line of equipment of major importance in his work, such as refrigerating equipment to be used for air conditioning. Consideration will also be given to free-lance contracting operations, where no arrangements exist for the exclusive purchase of equipment as offered by one manufacturer.

The unfranchised contractor-distributor asks himself, "What franchise best fits my needs?" This one question is of the utmost importance, and raises points which, surprisingly, have received little attention or analysis, at least in published form. It resolves itself fundamentally into a comparison of merchandising as against engineering-contracting.

To define the terms, consider merchandising as mass distribution of simplified, standardized, mass production articles and the engineering-contracting type of distribution will apply to individual, tailor-made assemblies of equipment, where almost no two jobs are handled just alike.

On the surface, there are many distributors and manufacturers who appear to be a combination of both the merchandising and the individual engineering types. It is the author's belief, however, that each organization takes quite definitely either one classification or the other, and that it is extremely rare that any organization *effectively* combines both types. Each organization has its habits of thought and action, its most effective and profitable field, along one line or the other for which it is better qualified by experience, reputation, and the capabilities of its personnel. It may venture into the opposite line in a considered step toward diversification, but it should not gamble a heavy investment on the need for an overnight change in habits of thought and ways of doing business. The change must be faced as a long pull, for it must permeate the whole organization.

Classifying manufacturers and distributors, then, as either of the merchandising or the engineering type, it is the author's next thought that the most effective combination is between a distributor and a manufacturer of like classification. They understand each other, and pull together.

It is a painful thing to see the results of a franchise connection between unlike types. Thus, a distributor of the merchandising type, franchised with a manufacturer of the engineering type, will hire an engineer, set up both a dealer and a retail sales organization, start a heavy local advertising campaign, call a dealer convention, and then find that his factory has none of the specialized field promotion men or the educational material required by his intense promotional campaign. His dealers bring in prospects, but his engineer has to do most of the work for them. The jobs that are being installed monopolize the engineer's time, and the rest of the organization is usually poorly fitted to handle the technical problems which arise during the engineer's absences. By the time the dealers are halfway trained, the distributor gets sick of it and changes to a line better suited to his needs, where he can use his merchandising skill. In the meantime, both he and the manufacturer have lost money, sales opportunities and prestige.

Where the engineering contractor lines up with a merchandising type of manufacturer, there are similar frictions and misunderstandings. To build up a dealer organization, and to manage a crew of retail salesmen, call for a special type of ability and experience which are not to be picked up easily by a typical engineering contractor. There are quotas, stocks, advertising arrangements, showrooms, drawing accounts, yearly model changes, sales conventions, synthetic enthusiasms, a free service period, special bookkeeping systems, and many other aspects of mass distribution which are well understood by the experienced but which are quite likely to lead the engineering contractor into confusion.

The Contractor-Distributor's Franchise Obligations

The merchandising type of franchise will usually call for a quota, sometimes consisting of an agreement to purchase a certain number of units during each month of the selling season. If the line is well advertised, has acceptance, and is being produced in large quantities for manufacturer's and distributor's inventories, the franchise must usually be accompanied by a wholesale stock order, often measured in carloads. A certain minimum sales organization is sometimes specified, together with one or more competent service men.

These requirements are all logical and fair, assuming that the quotas, stocks, etc., are figured within reason. Mass production, national advertising and a good public acceptance of the line all represent past and present investment by the manufacturer. The factory should not have to carry all of the gamble alone, and present a ready-made market to the distributor without any contribution on his part to balance the manufacturer's investments. The advertising and the past efforts and investment by the manufacturer have created the present public acceptance, which makes possible the mass production, which in turn gives the distributor and the public a standardized package job, at a saving in cost of production which may or may not be passed all the way along the line of distribution.

The franchise, whether it be of the merchandising or the engineering type, will stipulate that there must be one or more engineers in the contractor-distributor's organization capable of handling the applications of the equipment. Even the simplest of package units needs proper application, installation and service, while the engineering requirements of the tailor-made central plant are obvious. This is a proper measure of protection for the manufacturer, whose name-plate remains on the equipment, as well as for the distributor and the customer.

The contractor-distributor who is starting a franchise connection will be well advised to take only a moderate sized territory which he can readily cover. He needn't worry about getting his territory increased in a year or two, if he wants it, and has been doing a good job. In the meantime he will have less travel expense, and fewer arguments with his factory. Inquiries cost money, and the factory is bound to follow up the leads which it has forwarded to him, to make sure that the advertising has not been wasted, and that the prospect receives proper attention.

The merchandising type of franchise will usually have terms of payment calling for sight draft against bill of lading. The engineering franchise may have similar terms, or in some cases may offer open account where

the contractor has suitable credit standing. The document will have a clearly stated provision that payments are to be made when due, without regard to any possible counterclaims, returns under warranty, or other deductions not supported by credit memoranda. Liability for consequential damages is always denied by the manufacturer, and his liability is always carefully limited to replacement of defective parts for one year after shipment, at the factory, with others paying labor and transportation both ways.

There are occasional abuses of the open account terms by a contractor who may have had his collections held up by the customer's insistence that the equipment is not working to his satisfaction. There have also been cases where the contractor has held up payments to the factory in an effort to influence an allowance beyond warranty provisions, to cover labor and transportation costs, refrigerant losses, etc., allegedly caused by defective materials. These practices are not justifiable. The manufacturer has based his selling price, and thus his margin, on limited risks all of which should be under his control to a workable degree. He has no direct control over the sales methods and installation workmanship of the contractor who resells his equipment. If the manufacturer is to be asked to foot the bills for possible improper application of his equipment in the field, or for over-optimistic sales promises, or for careless installation workmanship, he must increase his selling price to the contractor by a corresponding margin. He would be foolish to do even this, because he cannot very well evaluate the risks of field applications by contractors not under his direct control.

Benefits of the Franchise to the Contractor-Distributor

The principal benefit of a franchise is the assurance that a line of equipment can be bought, for a considerable period of time ahead and with reasonable notice of change of status, at a known price level which is lower than prices for the same equipment to competing contractors.

Against this is the fact that the free-lance or unfranchised contractor, while penalized in the cost of that particular line of equipment, is free to shop around and buy whatever equipment he can find to meet the needs of the particular job. If he is willing to take the risks, he can always find something a little cheaper, especially where he scatters his orders for various items of equipment among several suppliers for individual price advantage. The free-lance contractor carries the responsibility of coordinating the many items properly, which may run into quite a service expense later.

Certainly the contractor will enter into a franchise only where he has full confidence in the line, both as to the quality of the equipment and as to the responsibility and reputation of the manufacturer. He should also be sure that the line to which he ties himself exclusively has a reasonably complete range of types and sizes, so that he will not be forced to bid equipment which is out of line in size and thus in price.

The franchised contractor receives, in regular routine, all leads and inquiries in his territory from the manufacturer, resulting from national advertising, from the activities of the manufacturer's national organization, and from his fellow-contractors for that line. The manufacturer sends bulletins, educational material, engineering reports, recommended layouts, literature, etc., and extends engineering and sales assistance by mail or wire or by visits of the manufacturer's representatives. These details of assistance and education are extended as a matter of course to franchised contractors. In general they are not available to free-lance resale customers. No manufacturer is going to take the trouble to educate an independent contractor in certain specialized tricks of engineering when he knows that this skill may be used against him on the next job.

Proficiency in air conditioning calls for specialized knowledge of a wide variety of equipment. The manufacturer should know his own line and its applications much more exhaustively than can any one contractor. Complete assistance, willingly given, should therefore be a strong consideration in the contractor's mind when he gives thought to taking a franchise. The free-lancer has to stand alone, dependent upon his own skill and resourcefulness, reaching what must be essentially a superficial knowledge of several lines of equipment, as compared with a complete and exhaustive knowledge of one line.

It must be admitted that most of those existing free-lance contracting organizations with which the author is acquainted are efficiently organized, very able technically, highly experienced, well financed and of excellent reputation. They have to be in order to make a success.

Direct Sales by Manufacturers

The manufacturer generally takes care of development and sales work with chain stores and other types of national customers. Some manufacturers have developed this department to a high degree. It may generally be said that the largest and most important chain store and national customers receive prices and discounts which make it difficult for the contractor-distributor to sell to them at a workable markup. Installation and service are sometimes arranged through the distributor organization for a nominal override on the equipment price plus a figure for the labor and miscellaneous materials involved. Frequently the national customer has its own engineering and installation forces.

In certain cases the contractor-distributor will be handed a sale "on a silver platter," where the manufacturer has made a blanket arrangement with a national customer which involves turning over the separate contracts to the local contractor-distributors for delivery and installation. The contractor should examine this kind of sales contract more carefully than if he had made the sale himself. He should consider the following points: first, with whom is he contracting, the user or the supplying manufacturer; second, are the terms of payment clearly stated and acceptable; third, is there anything unusual about the guarantee, or would it be difficult to meet its requirements under local conditions; fourth, why did the manufacturer not handle the contract himself,—is it a matter of policy, or is there something unusual about it?

It may seem ungrateful to be so fussy about taking a gift, but the very nature of the transaction sometimes

causes oversight of the primary rules of good business. No reputable manufacturer would knowingly shove onto his distributors any contracts which are questionable, but national customer sales are sometimes made in too much of a hurry, without due consideration for varying local conditions.

Relations with Competing Contractors

Tolerance is a mark, and conceivably a weakness, of the educated man. The highly skilled engineer is more likely to admit the good points of his competitor's position, than is the intense, half-trained salesman. It would seem that if a curve of business ethics and civilized behavior were plotted against various fields of endeavor, the field of engineering contracting would be found at one of the higher points of the curve, and high pressure merchandising would be found at a somewhat lower level.

Among those air conditioning contractors active on the largest applications, there is a very fine and ethical type of competition. It is decidedly unusual, for example, to hear any derogatory remarks about a competitor or his equipment. In general the atmosphere in this competitive field is decent, pleasant and friendly.

As the field approaches that of intense merchandising, we find a much less tolerant spirit. Here the selling is being done by semi-trained men, not necessarily of any particularly high educational background, who must depend upon their quick-wittedness and their emotional drive rather than upon a thorough mechanical or engineering knowledge of what they are selling and what the customer needs. Under pressure, the salesman will do and say things which he will regret later. The element of gambling is here present to a much greater degree. A merchandising organization can mushroom in a year to an impressive size and volume of sales, and it can go broke very quickly too. In such a field of competition there is very little room for the amenities.

From the viewpoint of the engineering type of contractor, it is a good thing to have a stable, prosperous group of competitors. Selling prices are likely to be at reasonable levels, and the installations are more likely to be of the highest type, where there is no pressing need to cheapen the layout. This happy state of affairs can conceivably result in unduly high prices to the customers, but if it continues too long there will always be a new price-cutting competitor coming in, attracted by the possibilities of the situation.

Conclusions

The air conditioning contractor must preserve a balance in his organization between the business and the technical sides. It is apparent that many technically able contractors have neglected certain rules of management, and it is for this reason that this paper is presented.

Among the outside business relationships of the air conditioning contractor, those with the customer and with manufacturing suppliers are frequently open to misunderstandings and consequent losses. Customer relationships are simplified when there is a fair and competent consulting engineer to supervise the job.

There is a fundamental difference between merchandising and engineering types of distribution, and the contractor-distributor should analyze his own organization, and that of a prospective franchise connection, carefully before lining up. A good franchise connection is shown to have many advantages, in spite of the obligations which attend it.

The competitive situation in the engineering-contracting field is much more tolerant than in the field of merchandising, with installations more firmly held to engineering standards.

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Humidity and Its Control, by Wallace H. Martin. *Ice and Refrigeration*, Vol. 96, No. 6, June 1939, pp. 515-516. Discussion of humidity conditions in cold storage rooms. Control of humidity with use of various types of refrigeration, blower systems and brine spray system.

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The Correct Use of Steam Traps, by J. Seddon Brundit. *Journal of the Institution of Heating and Ventilating Engineers*, Vol. 7, No. 77, pp. 260-267, (London). Discussion of traps and their use in a steam system. Description of four types of traps: (1) balanced pressure thermostatic trap; (2) liquid expansion thermostatic trap; (3) ball float trap with thermostatic air release; (4) inverted bucket trap.

•

Statistics on Fuel Used for Heating Apartment Houses in New York. *Heating and Ventilating*, Vol. 36, No. 7, pp. 60-61. Actual figures on fuel consumption compiled from a wide range of New York apartment houses and average results which may be used as guides in estimating the heating costs of similar buildings.

•

Mine Ventilation, by D. Hay Clive and I. C. F. Statham. *The Colliery Guardian* (London), Vol. CLIX, continued in four issues as follows: July 7, pp. 3-7; July 14, pp. 47-49; July 21, pp. 91-94; July 28, pp. 136-138. Paper taken from the tenth report of the Midland Institute Committee on Mine Ventilation, presented at the general meeting of the *Institution of Mining Engineers*. A review of present theory and practice of mine ventilation.

•

New York's First Completely Air Conditioned Apartment House, by Sullivan A. S. Patorno. *Heating and Ventilating*, Vol. 36, No. 7, July, 1939, pp. 28-31. Description of the complete air conditioning system in apartment house at 25 East 83rd Street, New York City. Individual air conditioning units are used with individual fresh air inlets for each apartment.

•

Drying Pulverized Coal, by B. J. Cross. *Combustion*, Vol. 11, No. 2, August 1939, pp. 25-27. Report of a series of experiments made to determine the equilibrium moisture content of four different coals under varying atmospheric conditions, as to temperature and humidity. Drying and hydration curves were plotted for each coal and the different sets of conditions tested.

•

Heat Transmission Coefficients, by A. F. Dufton. *Journal of the Institution of Heating and Ventilating Engineers*, (London), Vol. 7, No. 77, July 1939, pp. 221-245. See also, *The Heating and Ventilating Engineer*, (London), Vol. XIII, No. 145, July 1939, pp. 19-23. A plea for codification of fundamental data used by heating engineers. Discussion of available information on heat transmission and the desirability of making a table of heat transmission data for all types of building material.

•

The Warmth Factor in Comfort at Work, a Physiological Study of Heating and Ventilating, by T. Bedford. *Medical Research Council Industrial Health Research Board*, Report No. 76, 1936, (London). Bibliography, list of publications by this Board. A field investigation of the relations between comfort and environmental warmth. The investigation was made in factories and offices under ordinary working conditions and relates to the comfort of persons engaged in light work. Mainly young women and girls.

•

The Ventilation of Ships, by R. McDonald. *Journal of the Institution of Heating and Ventilating Engineers*, Vol. 7, No. 78, August 1939, pp. 272-295. Discussion of general principles of ship ventilation and British Board of Trade requirements for ventilation to insure comfort of third class passengers and crew. General discussion of heating and ventilating in various parts of a ship. Description of motors and fans and the special requirements for their installation to prevent noise and unusual wear on the motors. Duct system on board ship requires special construction, to accommodate the various needs of the many departments on ship. Air conditioning is used in the new large luxury ships in the dining saloons, lounges and some sections. Brief description of these systems.

•

Noise and Its Effect on Human Beings; Noise Control as a By-Product of Air Conditioning, by Carey P. McCord, Edwin E. Teal, and Wm. N. Witheridge. *American Medical Association Journal*, Vol. 110, No. 19, May 7, 1938, pp. 1553-1560. Discussion of present concepts of noise in its relation to the comfort and health of human beings. Definitions of sound, noise, intensity and loudness, decibel scale. Description of sound level meter, the audiometer and the tuning fork method of measuring the masking effect of noise. Effects of noise on animals and humans. Fatigue due to noise. Prevention and reduction of noise.

•

The Installation of Gas and Steam Equipment in a Municipal Kitchen, by Frederick Bell. *Journal of the Institution of Heating and Ventilating Engineers*, Vol. 7, No. 78, August 1939, pp. 296-317. Description of apparatus and arrangements to give a clean and easily operated kitchen, with a minimum of labor and attention. This kitchen was to be used to serve noon day meals to the necessitous children of a city, the meals to be delivered by trucks to the various schools. Meals for 10,000 children daily. Menus are given.

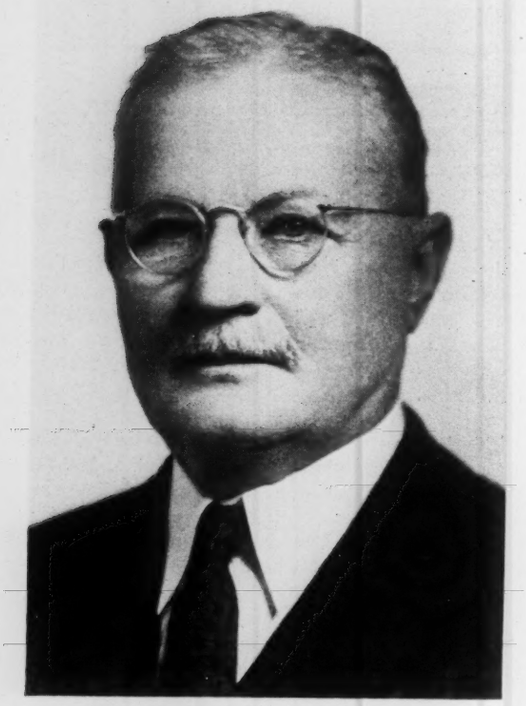
•

Steam Boiler Defects, by Sydney D. Scorer. *The Heating and Ventilating Engineer*, Vol. XIII, No. 147, September 1939, pp. 140-142. Discussion of the expected life of a steam boiler. Mention is made of some defects in boiler construction which may cause early replacement of the boiler. Type of fireboxes used is discussed and details of its construction which may cause it to fail to operate, are explained.

•

Combustion Made Easy, by Bill Maddock. *Power*, Vol. 83, No. 10, October 1939, pp. 628-29. Tables of complete data on most fuel components. The idea is to figure each component of the fuel as though it were being burned separately. Then the pound of air required to burn the fuel is the sum of the amounts necessary to burn each part.

Nominees for ASHVE Officers in 1940



F. E. Giesecke
College Station, Tex.
President

W. L. Fleisher
New York, N. Y.
First Vice-President

E. O. Eastwood
Seattle, Wash.
Second Vice-President

M. F. Blankin
Philadelphia, Pa.
Treasurer

Nominees for Council for 1940

Three-Year Term

J. F. S. Collins, Jr.
Pittsburgh, Pa.

E. N. McDonnell
Chicago, Ill.

T. H. Urdahl
Washington, D. C.

C.-E. A. Winslow
New Haven, Conn.

Nominees for Committee on Research

Three-Year Term

Photograph by Bachrach
Philip Drinker
Boston, Mass.

Axel Marin
Ann Arbor, Mich.

A. E. Stacey, Jr.
New York, N. Y.

J. H. Van Alsburg
Chicago, Ill.

J. H. Walker
Detroit, Mich.

ASA Ventilation Code

A meeting of the Sectional Committee for Ventilation Code was called by Chairman A. E. Stacey, Jr., October 25, in the Engineering Societies Building, New York City. Representatives of 45 organizations are participating in this work, which is under the auspices of *American Standards Association.*

While the scope of the Committee's work is rather extensive, the problem of immediate consideration is standards for air conditions for places of human occupancy, where the primary consideration is the control of temperature, humidity, air motion and odors, and also the control of air pollution or contamination not removable at the point of its production or issuance in industrial, manufacturing or construction operations.

It was decided to confine the present discussion to comfort air conditioning included in Item a of the Committee's scope, and the Code of Minimum Requirements for Comfort Air Conditioning developed and adopted by the ASHVE and the *ASRE* are used as a basis for discussion.

Those in attendance were: Frank H. Faust, *National Electrical Manufacturers Association;* W. E. Zieber, *American Society of Refrigerating Engineers;* Ernest Szekely, *National Association of Fan Manufacturers;* F. L. King, *National Fire Protection Association;* H. L. Miner, *Manufacturing Chemists' Association of the United States;* Dr. E. C. McCracken, *American Home Economics Association;* J. G. Skidmore, *Heating, Piping and Air Conditioning Contractors National Association;* Robert Close and T. H. Phelan, National Broadcasting Co., Inc.; Swen Kjaer, U. S. Department of Labor, Bureau of Labor Statistics; C. F. Brooks, Joint Refrigerating Industry Committee; and ASHVE representative A. E. Stacey, Jr., chairman, and by invitation T. M. Reed, Jr., and A. V. Hutchinson; and Cyril Ainsworth of the *American Standards Association.*

Management Association to Discuss Plant and Employee Efficiency

The *American Management Association* announces a 1939 Production Conference to be held at the Palmer House, Chicago, Ill., on November 15 and 16, the theme of the meeting being Industrial Preparedness.

A feature of the meeting will be a discussion of Modern Heating, Ventilating and Air Conditioning Applications and the introduction of the subject will be made by A. E. Stacey, Jr., vice-president, Buensod-Stacey Air Conditioning, Inc., New York, N. Y. At the special session devoted to the Effect of Air Conditioning on Employee Efficiency and Health, F. C. Houghten, director of the ASHVE Research Laboratory, Pittsburgh, Pa., will be a featured speaker and will give some results of recent research conducted under the auspices of the Society's Technical Advisory Committee on Air Conditioning in Industry.

The question, Are You Ready for Expansion, will be discussed by Raymond S. Perry, vice-president, Ingersoll Milling Machine Co.; the Effect of Building Design on Production Efficiency is the subject of Albert Kahn, president, Albert Kahn, Inc.; and Plant Engineering and Unit Cost Reduction will be the topic of H. K. Ferguson, president, H. K. Ferguson Co.

Solid Fuels Committee Meets

On Friday, October 6, a meeting of the ASHVE Technical Advisory Committee on Solid Fuels was held at the Battelle Memorial Institute, Columbus, Ohio, with R. A. Sherman, chairman, presiding and with the following members present: R. S. Dill, L. N. Hunter, P. Nicholls, H. J. Rose, E. T. Selig and, by invitation, Messrs. Cross, Johnson, Mulcey, Powell and Williams. Minutes of the previous meeting in April were read and approved and changes in the membership of the Committee since the last meeting were announced. Reports of various sub-committees were received and a discussion of plans for completing the work on three projects during the remainder of the year were discussed.

Preceding the meeting the committee were guests of the chairman at dinner in the Faculty Club of Ohio State University.

Cleveland to Have 46th Annual Meeting and 6th Exposition in January

CLEVELAND, Ohio, was selected by the Council for the Society's 46th Annual Meeting to be held January 22-25, 1940 and the Northern Ohio Chapter will act as host. Meeting headquarters will be at Hotel Statler located in the downtown business district and easily accessible to transportation lines, stores, theaters, and Lakeside Hall where the 6th International Heating and Ventilating Exposition will be held.

A preliminary schedule of events is announced by the Meetings Committee.

Tentative Schedule—46th Annual Meeting

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
Cleveland, Ohio—Hotel Statler—January 22-25, 1939

Monday, January 22

8:30 a. m. Registration—(Mezzanine Floor)—Hotel Statler
9:30 a. m. Conference of Chapter Delegates—Pine Room
10:00 a. m. Council Meeting—Tavern Room
10:30 a. m. Committee Meetings
2:00 p. m. Opening of 6th International Exposition—Lakeside Hall
2:30 p. m. Nominating Committee Meeting
10:00 p. m. Informal Get-together Party.

Tuesday, January 23

9:30 a. m. Technical Session—Reports and three papers—(Grand Ballroom)—Hotel Statler
2:00 p. m. Technical Session—Reports and three papers—(Grand Ballroom)—Hotel Statler
10:30 p. m. Informal Party sponsored by *NWAHACA* (Hollenden Hotel)

Wednesday, January 24

9:30 a. m. Technical Session—Reports and three papers—(Grand Ballroom)—Hotel Statler
2:00 p. m. Technical Session with *National Warm Air Heating and Air Conditioning Association*—Hollenden Hotel
7:00 p. m. Annual Banquet and Dance—(Grand Ballroom)—Hotel Statler
Toastmaster—Walter Klie

Thursday, January 25

10:00 a. m. Technical Session—Reports and three papers—(Euclid Room)—Hotel Statler
Installation of Officers
1:30 p. m. Council Meeting—Tavern Room

It is planned to devote Monday, January 22, to registration, Council and committee meetings, Conference of Chapter Delegates, and the opening of the Exposition.

On Tuesday two technical sessions will be held at the Hotel Statler and, on Wednesday, the morning technical session will be at the Hotel Statler and the afternoon session will be a joint meeting with the *National Warm Air Heating and Air Conditioning Association* at the Hollenden Hotel. The final technical session at which reports and papers will be given is scheduled for Thursday morning in Hotel Statler, at which time the new officers will be installed.

Chairman L. T. Avery of the Committee on Arrangements has appointed a group of sub-committees which are planning an entertainment program for members and their wives and guests, who attend the Cleveland meeting: G. L. Tuve, Honorary Chairman; Philip Cohen, Vice-Chairman, and L. S. Ries, Vice-Chairman; *Technical Sessions*, C. F. Eveleth, chairman; *Publicity*, John Paul Jones, chairman; *Finance*, E. W. Gray, chairman; *Inspection Trips*, C. A. McKeeman, chairman; *Attendance*, D. L. Tazel, chairman; *Banquet*, F. A. Kitchen, chairman; *Transportation*, A. L. Vanderhoof, chairman; *Entertainment*, H. E. Wetzel, chairman; *Registration-Reception*, Paul Gayman, chairman; and *Ladies*, Mrs. W. R. Rhoton, chairman.

General stimulation felt by all industry during the past few months may be responsible for increased activity in the leasing of space by exhibitors for the Sixth International Heating and Ventilating Exposition as less than 20 per cent of the total exhibit area is now available. Most of the exhibitors who have exhibited at previous Heating and Ventilating Expositions have engaged space for the 1940 event, and many new exhibitors have come in.

The Heating and Ventilating Exposition—the Air Conditioning Exposition—brings together, biennially, the leading manufacturers in this field. The Exposition represents the only time and place where so great a variety of air conditioning equipment and accessories can be examined and compared. Heating, ventilating, and refrigerating engineers, technologists in every classification of industry, contractors, builders, building owners and operators, public utility executives, jobbers and dealers come to the Exposition to see what is new and to make their individual comparison of values and claims.

The International Heating and Ventilating Exposition has enjoyed a remarkable record of success since the first one was held in Philadelphia in 1930. Subsequent presentations have been in Cleveland in 1932, New York, 1934, Chicago, 1936, and New York in 1938. Large attendance and phenomenal exhibitor representation have greeted each of these expositions and have offered evidence of a growing industry. Expansion of the industry during the past decade reflects many factors such as the movement toward better housing, improved environmental conditions for labor, progress in public health, and the advancement of engineering and science aided by organized research.

The Exposition deals with heating, ventilating and air conditioning in all possible applications. Admittance to the Exposition is restricted so that its values can be examined and enjoyed by the technologists and executives directly concerned. Admittance is by invitation or registration only.

MEMBERSHIP COOPERATION

The Fall season, the one of greatest activity in our Society, is here. This is the time when in many chapters new officers are taking over, new membership committees are starting work, and interest in the chapters and in the Society as a whole should be approaching the peak.

Interest in Society activities must be shown in more ways than by merely attending a meeting, eating a dinner, and listening to somebody speak. If members are really interested in the Society, they will talk about it to people who should also be interested. In this way, the best membership work can be done.

Let's not leave it all to the membership committee and to the new officers . . . let every member of every chapter, and those who have no chapter affiliation, talk about the Society—what it has accomplished and what it is accomplishing. Friends who ought to be members will then know that we are proud of the Society and its work. Many will be ready to listen and many will be ready to help.

E. K. CAMPBELL, *Chairman*,
Membership Committee, ASHVE.

ANNUAL REPORTS OF LOCAL CHAPTERS

Ontario

October 3, 1938. J. F. McIntire, Detroit, 1st vice-president of the Society, was the guest speaker at the opening meeting of the Ontario Chapter, which was attended by 56 members and their guests. Pres. McIntire chose as his subject for discussion The Engineer, which was greatly appreciated by the chapter.

November 7, 1938. This meeting was turned over to H. H. Angus, chairman of the Code Committee of the Chapter, who in turn, opened the meeting to discussion on the Code for the Design and Installation of Steam and Hot Water Heating Systems for Residences having up to 50,000 cu ft of volume, which was adopted by the Chapter with the changes recommended at this meeting following its presentation. There were 52 members and guests present.

December 5, 1938. The principal speaker of the evening was A. E. Stacey, Jr., New York, who gave an illustrated talk on Air Conditioning in Industry, which was followed by an interesting discussion. There were 58 in attendance at this meeting.

January 9, 1939. M. W. Shears, chairman of the Code Committee, proposed the adoption of the Warm Air Heating Code, which was adopted with the amendment suggested at this meeting, that the infiltration factors in this code agree with the infiltration factors in the Steam and Hot Water Heating Code recently adopted by the chapter. The guest speaker of the meeting was W. A. Duncan, who spoke on Modern Oxy-acteylene Welding of Piping, accompanied by a movie reel on welding.

February 6, 1939. There were 77 members and guests present at the February meeting held in the Royal York Hotel, Toronto. H. H. Angus, as chairman of the Open Forum on Domestic Hot Water Heating, outlined pertinent data pertaining to domestic hot water heating, and a film of various types of automatic control for domestic hot water heating was shown, followed by an open discussion.

March 6, 1939. J. H. Van Alsburg addressed the chapter on The Four D's of Air Conditioning in the presence of 69 members and guests. A vote of thanks was extended to Mr. Van Alsburg at the conclusion of his interesting talk.

April 3, 1939. The *April* meeting was attended by 85 members and their guests, who assembled to hear J. R. Hertzler, York, Pa., speak on Atmosphere and Efficiency in Its Engineering Aspects.

May 1, 1939. There were 39 members and guests present for the annual meeting of the chapter held at the Royal York Hotel, Toronto. All Committee chairmen reported on the past year's activities and the election of the following officers took place:

President—H. D. Henion.
Vice-President—J. W. O'Neill.
Secretary—H. R. Roth.
Board of Governors—H. B. Jenney, C. Tasker, F. E. Ellis and J. H. Fox.

Western Michigan

The first meeting of the season was held in *October 1938* at the Rowe Hotel in Grand Rapids with 45 members and guests in attendance to hear H. W. Ewald, the guest speaker of the evening, give an interesting presentation on air conditioning an auditorium.

The *November* meeting had as its speaker, W. S. Warnack, who spoke on the uses and manufacture of steel, before an attendance of 55 members and guests.

Air conditioning a glass brick building was presented by J. H. Walker at the *December* meeting, followed by an interesting discussion. There were 46 members and guests in attendance.

At the *January 1939* meeting Dr. C. A. Mills, professor of experimental medicine, University of Cincinnati, gave a talk on air conditioning for health, and a lively discussion followed. There were 15 members and 28 guests at this meeting.

A report and discussion of the papers presented at the Society's Annual Meeting in Pittsburgh were given at the *February* meeting by S. H. Downs, L. G. Miller, W. W. Bradfield and B. F. McLouth, all of whom attended the Annual Meeting. The attendance at this meeting was 42.

The *March* meeting was a joint meeting with the 8th Annual Forced Warm Air Conference at Michigan State College, East Lansing, where 225 gathered for the meeting including 20 members of the Western Michigan Chapter. There were informal talks by Dean H. B. Dirks, Prof. L. G. Miller, Prof. S. Konzo, F. R. Bishop, E. B. Root Prof. L. G. Hayes, C. A. Paquin, W. G. Boales and Sgt. Ray Sullivan of Michigan State Police, who gave an interesting sharp shooting exhibition.

The Nominating Committee reported at the *April* meeting and presented the names of the officers for the 1939-40 season. R. E. Seeger, city engineer, reviewed the suggested ordinance prepared by the National Board of Fire Underwriters and made several suggestions for revisions. The meeting was attended by 18 members and 14 guests.

The Western Michigan Chapter of the ASHVE closed their 1938-39 season with the May 1939 meeting at which they elected the following officers to serve for the coming year:

President—B. F. McLouth.
Vice-President—T. D. Stafford.
Secretary—C. H. Pesterfield.
Treasurer—C. H. Morton.
Board of Governors—C. R. McConner, W. G. Schlichting and F. C. Warren.

The committees appointed to serve are as follows:

Membership—C. R. McConner, chairman; R. E. Distel, W. F. Graff, S. J. Dempsey, R. W. Wilson, G. D. Woodhouse, R. L. Leigh and W. C. DeRoo.
Program—T. D. Stafford, chairman; L. G. Miller and H. G. Metzger.
Codes—W. G. Schlichting, chairman; L. G. Miller and S. L. Worthing.
Entertainment—W. W. Bradfield, chairman; V. H. Hill, H. J. Young.
Nominating—O. D. Marshall, chairman; S. H. Downs and H. F. Reid.
Publicity—F. C. Warren, chairman; N. C. Hotop and T. L. Osberger.

H. D. Henion
President

H. R. Roth
Secretary

B. F. McLouth
President

C. H. Pesterfield
Secretary

Massachusetts

The first meeting of the 1938-1939 season was held *October 17, 1938* at Walker Memorial, Massachusetts Institute of Technology. The guest speaker of the evening was Pres. E. Holt Gurney, who spoke at some length on the Society activities and later on Air Conditioning Brings Human Problems to the Engineer. Also present were Past Presidents, W. T. Jones and D. S. Boyden, and several of the officers of the Boston Air Conditioning Bureau.

The *November* meeting was also held at M. I. T. and had as guest speaker, Prof. Gordon B. Wilkes, who spoke on the general subject of Some Problems in the Use of Insulation. This proved to be a very timely topic as indicated by the discussion. Besides various types of insulation, the effect of various positions with reflective insulation, and the use of vapor barriers was discussed. There were 35 in attendance.

The *December* meeting had its Chapter president, James Holt presiding. The speaker was Prof. C. P. Yaglou, Harvard School of Public Health, who spoke on the Effect of Air Conditions on Comfort, Health, and Disease. This was an unusual talk and proved to be interesting to the 40 members and guests present.

At the *January* meeting F. J. Kurth, Anemostat Corp. of America, spoke on the general topic of Air Distribution and Diffusion. There were 90 members and guests in attendance.

Vice-Pres. R. M. Nee presided at the *February* meeting of the Chapter, at which Past President W. T. Jones reported on the Society's Annual Meeting in Pittsburgh. The speaker of the evening was A. E. Stacey, Jr., New York, who spoke on Air Conditioning in Industry. Mr. Stacey told of the work being done by this Committee of the Society in regard to measurements, correlation and tentative standards of the large amount of data already available from the health standpoint. The aims and proposed work of the committee were explained and discussed. At the close of an interesting discussion a rising vote of thanks was tendered Mr. Stacey.

J. H. Van Alsburg, Chicago, was the speaker at the *March* meeting and spoke on the Four D's of Air Conditioning, ducts, dampers, distribution and deflection. Mr. Van Alsburg not only covered the subjects of his talk, but also gave an interesting account of the projects being carried out in the various research laboratories in conjunction with the ASHVE. There were 50 members and guests in attendance, and it is understood that each went away with a better understanding of the importance of the work being undertaken.

The *April* meeting of the Massachusetts Chapter was in charge of R. M. Nee, Boston Edison Co., and vice-president of the Chapter. The subject of the evening was Steam Utilization—Engineering and Control, and a discussion was carried out by Mr. Nee and his assistants.

The final meeting of the season was held *May 23, 1939,* and concluded another successful year as guests of the Harvard School of Public Health. The speaker of the evening, A. L. O'Banion, engineer, Sumner Tunnel, City of Boston, discussed the ventilation and operation of the Sumner Tunnel, as well as its construction. The many questions asked at the conclusion were answered in an interesting manner by Mr. O'Banion.

The officers elected to serve the Massachusetts Chapter for the 1939-1940 season are as follows:

President—H. C. Moore.
Vice-President—C. P. Yaglou.
Secretary-Treasurer—C. M. F. Peterson.
Board of Governors—D. S. Boyden, D. J. Edwards, James Holt, Alfred Kellogg, R. M. Nee, H. M. Tarr and J. F. Tuttle.

New York

The New York Chapter opened its activities with the *October 17* meeting with Pres. E. Holt Gurney as the guest speaker. President Gurney gave an interesting address on Air Conditioning Brings Human Problems to the Engineer.

The *November 21* meeting, held at the Building Trades Club, had 110 members and guests in attendance. The Chapter was fortunate in having Alfred L. Jaros and A. C. Buensod as guest speakers at this meeting. Mr. Jaros spoke on the design and supervision of comfort cooling and ventilation of Longchamps Empire State Restaurant, and Mr. Buensod selected the fundamentals required for chain store cooling as his topic.

The *December* meeting, with an attendance of 181 members and guests, was in the form of a Christmas party which was the fifth annual one held by the New York Chapter.

Dr. J. H. Kimball, meteorologist of the New York Weather Bureau, gave an interesting talk on the history and development of the recent hurricane at the *January* meeting. Another interesting talk at this meeting was on the theory and practice of air distribution in air conditioning systems, which was given by F. J. Kurth, Anemostat Corporation of America. The meeting was well attended, having 235 members and guests present.

The speakers at the *February* meeting of the Chapter were J. R. Parsons, who spoke on effective temperature, and L. N. Hunter, who addressed the meeting on the uniform rating of cast-iron house heating boilers as it will be used by *The Institute of Boiler and Radiator Manufacturers.* There were 69 in attendance at this meeting.

The *March* meeting was given over to members of the New York World's Fair staff who covered the heating, ventilating, air conditioning and other features of the Fair in their more interesting aspects. The speakers were, Joseph Judge, coordinator of the Board of Design at the Fair, H. W. Durham, Construction Department, and Sanford R. Apt, chief mechanical engineer and member of the ASHVE.

Dr. C. W. Brabbée discussed 40 years of heating, ventilation and air conditioning at the *April* meeting, followed by C. E. Mayette, who presented the trend of heating, ventilation and air conditioning. Mr. Mayette was formerly principal mechanical engineer for the U. S. Housing Authority at Washington. The meeting was attended by 77 members and guests.

On April 26, the members of the New York Chapter visited the glass exhibit at the World's Fair, courtesy of Glass Inc., and were entertained at luncheon immediately after at Ballantine's on the Fair grounds.

The annual *May* party, held at the Building Trades Club with 197 in attendance, closed the 1938-39 season of the New York Chapter, with the election of the following officers for 1939-40:

President—O. O. Oaks.
Vice-President—C. S. Pabst.
Secretary—T. W. Reynolds.
Treasurer—W. M. Heebner.
Board of Governors—H. G. Meinke, H. L. Baker, Jr., and A. L. Browne.

Philadelphia

The Philadelphia Chapter opened its 1938-39 season with the meeting on *October 13, 1938.* Dinner was served to 69 members and guests at the Engineers Club, and E. Holt Gurney, then president of the Society, was the guest speaker. The guest of honor was T. Harold Fox, British Vice Consul.

There were 77 members and guests at the meeting on *November 10* when G. L. Wiggs, Montreal, presented a talk on modern hot water heating practice.

Prof. W. F. Wells gave an interesting discussion on sanitary ventilation at the *December* meeting, which was largely attended. A lively and constructive discussion followed Professor Wells' talk.

The *January* meeting was held at Kugler's restaurant, where dinner was served to 104 members and guests. With the exception of the reading of the minutes, all other regular business was suspended by Pres. H. H. Erickson, to permit time for discussion on the recommended changes in the By-Laws.

February 9 was the date of the next meeting of the Chapter, held at the Engineers Club with an attendance of 51 members

and guests. The guest speaker, R. E. Daly, spoke on modern boiler design.

H. J. Knowles, American Air Filter Co., discussed recent developments in air cleaning apparatus at the *March* meeting which was held at the Engineers Club.

The *April* meeting had as its guest speaker W. A. Grant, Carrier Corporation, who gave an entertaining talk on zoning of air conditioning systems.

The *May* meeting was the annual golf outing which brought an attendance of 63 members and guests to the St. Davids Country Club, Wayne, Pa., where the following officers were installed for the coming year:

President—R. F. Hunger.
First Vice-President—C. B. Eastman.
Second Vice-President—H. B. Hedges.
Secretary—Edwin Elliot.
Treasurer—H. H. Mather.
Board of Governors—R. F. Hunger, H. H. Erickson, C. B. Eastman, H. B. Hedges, H. H. Mather, Edwin Elliot and M. G. Kershaw.

The personnel of the various committees to serve for the coming season are as follows:

Meetings—H. B. Hedges, chairman; A. C. Caldwell, L. A. Childs, G. E. Tuckerman and C. S. Leopold.
Membership—C. B. Eastman, chairman; F. D. Mensing, J. H. Hucker, J. W. McElgin, W. J. Searle, Jr., R. J. Steel, M. E. Barnard and E. K. Wagner.
Legislative—H. H. Erickson, chairman; L. P. Hynes and W. P. Culbert.
Finance—H. H. Mather, chairman; W. F. Smith, L. P. Hynes, M. F. Blankin and H. H. Erickson.

Buenger Joins A. M. Kinney, Inc.

A. M. Kinney, president of A. M. Kinney, Inc., consulting engineers, Cincinnati, Ohio, announces that Albert Buenger has joined the staff, bringing with him a wide experience in general practice and specialized knowledge in the air conditioning field. For 16 years he was associated with C. H. Johnston, architect, St. Paul, as mechanical engineer, after which he spent three years with the Delco-Frigidaire Conditioning Division, General Motors Sales Corp., Dayton.

Mr. Buenger, a member of the ASHVE since 1917, served on the Society's Council from 1934-1937, on the Committee on Research for 1934 and 1935, as chairman of the Guide Publication Committee for 1937 and 1938, and other Society Committees. He also served the Minnesota Chapter as chapter president, and also as secretary-treasurer.

New Consulting Service

N. Bert Persson, M.E., announces a consulting service for restaurants, hotels, hospitals and institutions under the name of Food Service Equipment Engineering, 1418 Simpson Ave., St. Paul, Minn. Mr. Persson will specialize in designing, air conditioning, commercial refrigeration and ventilation systems.

Col. W. A. Danielson

Washington, D. C. Honors Colonel Danielson

September 20, 1939. Col. W. A. Danielson, first President of the Washington, D. C. Chapter, was honored at a social meeting of the Chapter at the Washington Golf and Country Club. R. H. Feltwell, on behalf of the ten organizing members, presented the Chapter with a rosewood gavel with a silver band containing the following inscription:

In Honor of
Colonel Wilmot A. Danielson
Founder and First President 1935-37
Washington, D. C. Chapter
American Society of Heating and Ventilating Engineers

Due to his recent transfer from Fort Knox to the Canal Zone, Colonel Danielson was unable to be present, and the gavel was received by T. H. Urdahl, President of the Chapter, and is to replace the very battered tinner's mallet which had served as a gavel in past years. C. V. Haynes, past president of the Society and a member of the Washington, D. C. Chapter, offered several remarks most appropriate to the occasion and also appropriated the worn mallet. Just what sentiment this antique holds for him is not known, but it is more than a presumption that "Charlie" will attach a long tale to it when he goes banging around to the other Chapters.

There were 65 members and guests present, and more than half kept Mr. Haynes company on the golf course during the afternoon. It is now a known fact that golfers in this Chapter are not only numerous, but able. Ten new members of the Society were introduced, including F. B. (Chick) Sale, of Preferred Utilities, who is confining his efforts to accessories only.

The golf outing and entertainment were arranged and directed by W. E. Kingswell, Minneapolis-Honeywell Regulator Co., to whose remarkable control over the golf ball and his Committee, much of the success of the meeting is due. Following the meeting auditions for candidates for the Chapter Glee Club were held by M. F. Hoppe, Chairman of the Meetings Committee, who looks forward to greeting all comers to the summer meeting in 1940.

CANDIDATES FOR MEMBERSHIP

The Constitution of the Society, as now amended, requires the following mode of procedure in voting on applicants for membership in the Society. All applications for membership are to be sent to the Secretary and the names of applicants and their references shall be printed in the next issue of the JOURNAL of the Society or sent to the members in other approved manner as ordered by the Council. When replies are received from references, the Candidate's application shall be submitted to and acted upon by the Committee on Admission and Advancement as soon as possible.

When the Committee on Admission and Advancement has acted favorably upon a Candidate's application and assigned his grade, the Council shall vote upon the election of the proposed Candidate for membership by letter ballot. During the past month 20 applications for membership have been received and the names of these men and their sponsors are published in the following list.

Members are requested to scrutinize the list with care. The Committee on Admission and Advancement, and in turn, the Council, urge the members to assume their share of responsibility of receiving these candidates into membership by advising the Secretary promptly of any whose eligibility for membership is in any way questioned.

All correspondence in regard to such matters is strictly confidential, and is solely for the good of the Society, which it is the duty of every member to promote.